THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 34 FEBRUARY 1912 NUMBER 2

ISSUES OF THE JOURNAL WANTED

The Society wishes to secure copies of the following issues of The Journal: Mid-November 1906, Mid-October 1907, May 1908, October 1909, October 1910, January 1911. These will be purchased from members at 25 cents a copy, provided they are in good condition.

COMING MEETINGS

NEW YORK MEETING, FEBRUARY 13

The American Institute of Mining Engineers will coöperate in a meeting of the Society in New York, February 13, in the Engineering Societies Building. A paper of interest to both branches of the profession, entitled, Notes on Design and Mechanical Features of the California Gold Dredge, by Robert E. Cranston, consulting engineer for the Marysville Dredging Company, Marigold, Cal., will be presented. The author has made a careful study of the subject and the paper is fully illustrated with diagrams and data, drawn from dredges in use in California.

The paper will be discussed by Charles Whiting Baker, Editor of Engineering News; and by B. B. Lawrence, Julius A. Lewisohn and A. P. Rogers, Members A. I. M. E., of New York; and others.

It is anticipated that this paper will also be discussed at a meeting of the Society in San Francisco at an early date, and that the viewpoints brought out at these two meetings should prove to be a valuable symposium.

BOSTON MEETING, FEBRUARY 16

A meeting of the Society will be held in Boston on February 16. A topical discussion on Aviation will occupy the evening, the opening paper being presented by A. A. Merrill.

ST. LOUIS MEETING, FEBRUARY 7

A meeting of the Society will be held in St. Louis on February 7, in which the other engineering societies will be invited to coöperate. A paper will be presented entitled, A Comparison of Air Braking Systems for Urban Electric Railway Cars, by Messrs. Hobein and Young of the United Railways Company of St. Louis.

NEW COMMITTEES IN BOSTON AND NEW YORK

At a meeting of the members of the Society resident in Boston and vicinity, on January 16, the following committee were chosen to act as the Committee on Meetings in that city for the current year: E. F. Miller, Chairman; R. E. Curtis, Secretary; R. H. Rice, Henry Bartlett, George F. Swain. This committee has been approved by the Council.

H. R. Cobleigh has been elected upon the New York Committee on Meetings, to take the place of Walter Rautenstrauch, who has been obliged to resign because of other duties.

REPORTS OF MEETINGS

ENGINEERS' DINNER IN BOSTON

The third of a series of annual engineering dinners took place in Boston, on January 15, at the Hotel Somerset. The responsibility for the arrangements rested this year with the Boston Society of Civil Engineers, the Boston Section of the American Institute of Electrical Engineers and The American Society of Mechanical Engineers coöperating in the plans. There was an attendance of about 400 engineers from the locality of Boston, and a spirit of cordiality was everywhere in evidence.

Charles T. Main, Mem. Am. Soc. M. E., President of the Boston Society of Civil Engineers, presided, and in his opening remarks referred to the important and influential position of the engineering profession and spoke of the value of social intercourse between its members.

Ira N. Hollis, Mem. Am. Soc. M. E., president of the new Engineers' Club of Boston, was then introduced. He spoke of the neces-

sity of the emphasizing and fostering of unity throughout the engineering profession by intimate contact between its various groups. He told also of the progress made by the new club which had been incorporated and had chosen a strong Board of Governors. An option had been secured upon some property on Commonwealth Avenue facing the public gardens, an unsurpassed location for convenience and attractiveness of surroundings, which would be suitably altered to provide the usual facilities, and also an auditorium sufficient for ordinary gatherings, as well as quarters for the office and library of the Boston Society of Civil Engineers. It seemed probable that the full quota of members permitted by the By-Laws would soon be secured.

General Hugh Bancroft, chairman of the recently appointed Directors of the Port of Boston, was the next speaker, and dwelt particularly on the work confronting the directors. Large and important as might be the problem of providing new docks and terminal facilities, the development of business to utilize them was even greater. The opening of the Panama Canal, he prophesied, would make ports situated like Boston of more value than inland centers, and such a condition should insure the local support of enterprises tending to develop commerce at this port. He recommended that the water front should be the property of the public rather than of private individuals, in order to utilize the harbor facilities properly, and referred in conclusion to the intimate relation between the adequate development of the port and the industrial growth and prosperity of the state.

President Main then called upon representatives of the three local technical schools, emphasizing the close relation between these schools and the industrial and engineering interests of the locality.

Prof. W. C. Sabin, Dean of the Graduate School of Applied Science of Harvard University, spoke of the necessity for scientific training to keep in touch with the problems of the day, and referred to the experimental work with large scale models which had been done on the continent of Europe in connection with the study of river and harbor problems, and suggested the development of similar courses and laboratories in this country, so that such engineering work might be based upon a definite action of tides, currents, etc., with relation to the character of soil, conformation of shores, etc., substituting thereby scientific for empirical methods.

Dr. Richard C. Maclaurin, President of Massachusetts Institute of Technology, said that the institute also, like the new club, found

itself confronted with a housing problem, which it hoped to work out in an efficient and worthy manner. He referred to a suggestion which had been made that the institute model its new buildings after factory construction, and argued that advantage should be taken to the utmost of types of construction specially developed and proved to be satisfactory for industrial purposes, with the proviso that regard should also be paid to beauty of outline and architectural treatment, in order that the buildings should have a dignity which might suitably express the work and position of the institute in the community.

President F. W. Hamilton of Tufts College said that although as a non-technical man he might appear at a disadvantage in addressing a gathering of engineers, the fundamental principle of education was nevertheless the same, that of teaching the individual to think clearly and definitely and enabling him to attack his life's work with credit to himself and service to his community.

The greetings of the national engineering societies were extended by Admiral M. T. Endicott, President of the American Society of Civil Engineers; Colonel E. D. Meier, Past-President and Honorary Vice-President of The American Society of Mechanical Engineers; and Dr. Gano S. Dunn, President of the American Institute of Electrical Engineers, each of whom cordially indorsed the gathering together of engineers for intercourse and urged greater participation in public matters where they might serve the community.

NEW YORK MEETING, JANUARY 9, 1912

The January meeting of the Society in New York was unique among such meetings, and for that reason one of the most interesting that has been held. It was a "get-together" meeting for a statement of plans and policies by members of the various standing committees and of the Committee on Meetings in New York, and for a free discussion of these policies from the floor.

The Society is characterized at the present time by marked activity all along the line on the part of the various committees, and by the individual interest of many of its members. It was to acquaint the local membership with these developments and further to enlist their interest that this meeting was called.

The meeting was opened by Professor Walter Rautenstrauch, of the Committee on Meetings in New York. He introduced the President, Dr. Alexander Humphreys, who made the opening address and afterwards presided.

Dr. Humphreys took as his text a remark by Professor Rautenstrauch to the effect that ten per cent of the success of a project is due to its inception, while ninety per cent depends upon putting it into effect. He was in accord with the idea of making the Society more useful, but the question was, how can it be done thoroughly? There are obligations to be met, as for example the retirement of the certificates recently issued to the membership, provision for which has to be made out of the income of the Society. He favored the extension of meetings and discussed the possibility of the appointment of sub-committees to cover different activities in the field of mechanical engineering, as recently proposed by the Committee on Meetings, but said we should undertake only such work as we can do thoroughly, and at least the work should be planned with the evidence of careful thought.

With respect to the conduct of meetings, the Annual and Spring Meetings of the Society are under the control of the general Committee on Meetings, while the local meetings in different cities are under the direct control of the Council, this arrangement having been decided upon because these latter meetings are a new feature in the activities of the Society, and the Council does not yet know how to instruct the Committee on Meetings as to policy and method of management.

Dr. Humphreys discussed in detail various plans that had been proposed for raising funds for the entertainment at the Annual Meeting. At present this is done by subscriptions solicited from the membership in the New York district, and the Committee on Meetings in New York recently suggested an amendment to the Constitution to the effect that there should be a separate organization of the New York members in respect to annual dues and that every member within this district should pay an extra fee. The Council did not think this plan would be fair, as a man living here might be so situated that he could not take advantage of the local benefits and still he would be charged with this additional sum. He suggested that if it is still to be done by subscription, we would get better results if the amount asked for were limited to a specified sum, thus attempting to place the burden on a considerable number instead of on a few.

In the American Gas Light Association the difficulty is overcome by asking every member in advance of the meeting whether he is going to attend and for how many guests he will wish to provide.

Tickets are then provided him for the entertainment desired, and he is charged with a corresponding sum.

Dr. Chas. E. Lucke of the Committee on Meetings followed, outlining the functions of the Committee on Meetings during the past two years. Originally this committee had the double duty of the solicitation and judging of professional papers, and the management of all meetings.

At that time meetings were held only in New York, aside from the Annual and Spring conventions. With the extension of meetings to other cities, however, the problem of management became more difficult. Restrictions imposed by the officers of the Society on the conduct of meetings were in some cases objected to. Members in one section felt they were as well able to uphold the standards of the Society as were members in another section. The Council finally took the management of these local meetings entirely out of the hands of the Committee on Meetings and began by removing all restrictions, to which the speaker objected on the ground that while each city might be able to maintain a standard of its own, the standards in different localities differed.

Having been relieved of these responsibilities, the committee devoted more attention to its other work, the solicitation of papers. Investigation made it plain that the Transactions of the Society were not reflecting the practice of mechanical engineering broadly. It appeared clear that the Committee on Meetings, of five ordinary members, were not competent to judge all things in mechanical engineering. A paper had seldom been turned down that did not raise a storm. Ocassionally an expert was called in, but he did not like to be criticised either, and it was finally decided to recommend a series of sub-committees, so that if a man presented a paper that dealt with the engineering problems of the textile industry for example, he would know that it would be submitted to a committee of men representative of that industry. But why limit these committees to mere reference duties when they might be more valuable to the Society if, instead of merely asking them to tell what is good or bad among the papers submitted, they were to take the initiative and say what should or should not be treated at our meetings? It was with these ideas that the committee proposed to the Council to have a large number of sub-committees appointed. The membership was asked for suggestions of names for the committees and they are now being tabulated, but cannot be announced until the Council has determined upon a plan for their management.

Frederick A. Waldron, in speaking for the New York Committee, asked if it would not be advisable to make this a continuing committee in rotation, one member retiring and a new member appointed each year. One of the important problems is the raising of enthusiasm and interest on the part of the 1100 local members, in order to insure a reasonable attendance at the monthly meetings in New York. Boston meetings have a higher average attendance than New York meetings. There are no doubt ample reasons for this, owing to the peculiar conditions existing in New York, such as specialization of engineering work and the distance of the suburbs from the center of the city.

In respect to finances, which are so important an element in the committee's work, money must be obtained for two distinct purposes. First, there is an appropriation by the Society for defraying the expenses of the monthly meetings, which must be conducted within the limits required by the Constitution and By-Laws. These preclude the expenditure of money for social entertainment. Under the present arrangement money must be raised by subscription by this committee for the purpose of entertaining the visiting guests of the Society at the Annual Meeting. Various facts relating to the raising of this sum and to its expenditure (pertinent only to the local membership) were discussed by the speaker, who mentioned five ways by which the fund could be raised, all of which, he said, were open to more or less objection. These are as follows: (*a*) By increasing annual dues of local members; (*b*) by voluntary subscription on the part of local members; (*c*) by an underwriting organization in which prominent members of the Society would underwrite the expenses for the social features and obtain the money by whatever method the organization might see fit, making it really a permanent organization of the Society; (*d*) by the classification of business houses in the city from whom subscriptions would be solicited, no house to be called upon to subscribe oftener than once in three years; (*e*) by the pay-for-what-you-get method for both local and visiting members, which while attractive from the financial side cannot appeal to one strongly on the score of hospitality.

Fred R. Low, for the Publication Committee, explained that in the earlier days it was possible to include in the annual volume of Transactions everything said and done at the various meetings. This is no longer possible. The last volume, after drastic editing and the exclusion of much really pertinent matter, contained over fifteen hundred pages, which is about the limit both as to bulk and

expenditure. It is the purpose to make this annual volume of Transactions of the utmost possible permanent value and the test applied in discarding surplus material should be its probable usefulness to the man who gives it place on his shelves. Measured by this scale, it is possible that records of business sessions, reports dealing with the internal affairs of the Society, which are of no technical value and of only temporary interest, might find all the necessary publicity in the columns of the monthly Journal.

In The Journal, papers to be presented at the regular meetings are printed, most of them considerably in advance of the time of presentation, affording an opportunity for more deliberate and mature discussion. The Society has adopted the broad and liberal policy toward the technical press of releasing this material for publication before it has been formally presented at a meeting. Some of the societies which have publications of their own will not allow the technical press to print their papers or report their discussions until they have been given at public meetings.

But even in The Journal, it is not possible to print all of the papers presented at the various local meetings in the different cities, nor can all of the discussions given at these meetings be published. Notwithstanding the insufficiency of space for the complete presentation of our doings, the Publication Committee have initiated a movement for keeping readers in touch with the activities of the old world in the field of mechanical engineering. Foreign periodical literature coming to the library is gone over by a competent person and abstracts of articles made for publication in The Journal. Through the liberal patronage of its advertising pages The Journal is now self-supporting and the Society has again shown its breadth of view and appreciation of its real objects by offering it to the public practically at cost. An effort has been made to give a distinctive character to its advertising pages, in keeping with that of the Society, by the presentation of data valuable to engineers, regarding the apparatus advertised. Some three hundred pages of such material was secured last year in addition to the ordinary display advertising, and these are shortly to be issued in a separate volume for distribution to the membership.

Hosea Webster for the Membership Committee said that its chief work was the scrutinizing carefully of the qualifications of those who applied for membership. He suggested several ways in which lost motion might be eliminated and the nature of the requirements be

made more explicit, but was not prepared to make definite recommendations.

Dr. Leonard Waldo for the Library Committee, referred to the growth of the library of the Engineering Societies since it has been installed in its present location and of the many difficulties to be overcome. He spoke of his visit to various libraries in Germany during the past two summers and in particular of the excellent library of the Krupp works, where they have five or six hundred assistants on the library staff and hand out over two thousand books a day. He explained the desirability for coöperation between our library and the Astor library in the New York Public Library building almost across the way and that some of the expensive reference works could better be placed there than here. Every courtesy is shown those wishing to investigate engineering matters, and engineers who have become accustomed to consulting our library never wish to go elsewhere for such purpose. We are making systematic lists of fine books and as rapidly as the money can be secured it is being invested in that direction. We mean to have it so that when a request comes from any source for engineering information, the person will be able to receive the last word on the subject, whether he makes the request in person or in writing from a distance.

Edward Van Winkle, for the House Committee, outlined its work, beginning in 1907 by re-arranging the rooms, supplying new furniture and furnishings to a considerable extent, and in systematically grouping the portraits on the walls. The plan, as now carried out, is to hang the portraits of prominent engineers on the walls of the Council room, which is the first room entered upon leaving the elevator. The second room, used as a ready reference library, was set aside for a set of pictures of past-presidents, and the third room, the Secretary's office, for pictures of Honorary Members. It has been the aim of the committee to give the decoration of the rooms a personal touch such as would be given in the home.

A written statement was read by the Chairman of the Finance Committee, R. M. Dixon, showing the assets of the Society to be approximately $700,000, due largely to the generosity of Mr. Carnegie in his gift of the Engineering Societies building. The Society has no indebtedness aside from its funded debt of $81,000 represented by certificates of indebtedness which are to be retired year by year from the income. In 1900, the income of the Society was approximately $30,000. In 1911 it was approximately $97,000, and during the eleven years the expenditures with one exception have

been held within the income. At the beginning of each year, the Finance Committee presents to the Council a budget containing the appropriations for the various activities for the ensuing year. The present Finance Committee is of the opinion that it is none too conservative to base this budget upon a ten per cent margin of safety under the estimated income, and the appropriations for the current year have accordingly been made on this basis.

During the meeting there was considerable general discussion from the floor mostly upon various methods for raising funds for defraying the expenses of the entertainment at the annual meetings. There was the usual diversity of opinion upon the subject, different members favoring certain phases of the plan suggested by Mr. Waldron in his address on this subject, the sentiment being divided between increasing the dues of the local membership and continuing the present policy of soliciting subscriptions annually.

MEETING OF THE COUNCIL

A meeting of the Council was held in the Society rooms, on the afternoon of Tuesday, January 9, 1912. There were present: Alex. C. Humphreys, President, presiding, Ira N. Hollis, Jesse M. Smith, James Hartness, Henry Hess, Stanley G. Flagg, E. D. Meier, George A. Orrok, H. G. Reist, R. M. Dixon, Chairman of the Finance Committee, and the Office Manager representing the Secretary. Dr. Waldo, Chairman of the Library Committee, was present by invitation. Letters of regret were received from H. H. Vaughan, E. B. Katte and D. F. Crawford.

Voted: To approve the minutes of the meeting of December 8, 1911, as read.

The Secretary reported the death of Warren S. Johnson of Milwaukee.

Voted: To accept the following resignations: F. H. Niles, A. J. Hill, George G. Hopkins, Jr., F. J. Hale, James U. Norris.

The President made announcement of the following appointments on Standing and Special Committees: House, W. N. Dickinson; Library, Leonard Waldo (reappointed); Meetings, H. E. Longwell and H. L. Gantt; Membership, H. C. Meyer, Jr.; Public Relations, Worcester R. Warner; Research, A. L. DeLeeuw, Richard H. Rice (reappointed); Committee on Cooperation with Other Societies, E. D. Meier, C. W. Baker.

The Secretary reported the election of an Executive Committee by letter ballot, in accordance with instruction of the Council, to consist of Alex. C. Humphreys, President, ex-officio, F. R. Hutton, E. D. Meier, George A. Orrok, H. G. Reist and H. G. Stott.

The President presented a letter of resignation from H. G. Stott.

Voted: That Mr. Stott's resignation as a member of the various committees on which he is serving be accepted with great regret, but that he be requested to allow his resignation as Manager to be withdrawn.

Voted: That E. B. Katte, standing next in the order of votes received, be appointed a member of the Executive Committee to serve in the place of H. G. Stott whose resignation has been presented.

Voted: To approve and confirm the appointment by the Boston membership of E. F. Miller, R. E. Curtis, Richard H. Rice, Henry Bartlett, and George F. Swain, as a Committee on Meetings of the Society in Boston.

Voted: To refer to our Committee on Coöperation with Other Societies, the communication of Mr. Herbert E. Ives, Chairman of the Committee on Reciprocal Relations of the Illuminating Engineering Society.

A report from H. G. Stott, Chairman of the Committee on Revision of the Standard Schedule of Flanges, was read.

Voted: To adopt the recommendations of the Chairman that the report be printed in The Journal and leaflet form for distribution and that the Committee be discharged with thanks.

Voted: To refer to the Executive Committee with request to report back to the Council, the question of the status of the Committee on Constitution and By-Laws and that the Executive Committee ask the coöperation of Mr. Smith.

Voted: That regular meetings of the Council be held on the second Tuesday of each month at three o'clock in the afternoon.

Voted: That during the absence of the President and Treasurer, Ira N. Hollis, Vice-President, senior by age, be authorized to sign checks.

Voted: That it be referred to a committee to advise in the matter of a definite amendment to the Constitution which will define strictly the duties of the various officers of the Council and the assignment of their duties when for various causes they are not available.

On motion adjourned to Tuesday, February 13.

CURRENT AFFAIRS OF THE SOCIETY

That the members of the Society living at a distance from headquarters might feel that their interests were of equal importance to the officers and council of the Society, the Council sent the Secretary to San Francisco to attend a conference of local representatives from nine national societies, called to consider the advisability of holding a general engineering congress in 1915. The date of the proposed congress has been selected with reference to the opening of the Panama Canal, a great engineering achievement.

On his way to attend the conference the Secretary visited many of the Student Branches, including the University of Cincinnati, Ohio State University, the University of Kentucky, Washington University, the University of Missouri, the University of Nebraska, and the University of Illinois, and met the membership in many of the principal cities en route.

In Buffalo, on January 3, the first meeting ever held in that city under the auspices of the Society brought out an attendance of 200, representing different branches of the profession. In addition to the address by the Secretary, a paper was read by John Calder on the Economic Administration of Industrial Establishments, in which he spoke of the need for greater efficiency, and said that the organization of such an establishment was the primary consideration rather than the system, which should be used as a means, not as an end.

Members of the Society also met to greet the Secretary in Cincinnati, on January 4, and in St. Louis an informal dinner and reception were given on January 6. Similar meetings were held in San Francisco, Denver, and Chicago. At the Chicago meeting an address was made by R. H. Hunt, Past-President.

The benefits, to students on the one hand, of the student engineering societies, and the duty, on the other hand, of the national societies to interest themselves in the work of educating engineers need not even be stated.

With the older members of the Society the good to be secured

from meetings needs to be repeatedly brought out. Because of many demands, it often happens that a member may have been associated with a society for a dozen years without ever having attended a meeting or knowing the other members living in his own city.

At these meetings the Secretary has been able to present the many activities of the Society and mention its very practical benefits, such as those from the large technical library and its staff of trained workers, ready to make searches in any engineering subject. For simple searches requiring only bibliographies of articles on subjects appearing in current technical periodicals, 700 of which are received in the library, and of titles of books in the library, no charge is made. This service is free alike to the members of the American Institute of Mining Engineers, the American Institute of Electrical Engineers, and The American Society of Mechanical Engineers, whose libraries are combined in the United Engineering Library.

In our own Society there is another service at the disposal of the membership, that of the employment bureau. Firms desiring the services of competent engineers in any line may obtain a list of such engineers; and, vice versa, members of the Society desiring to improve their situation or others seeking a position may often secure opportunities. This service is absolutely free and is confidential, the work being personally conducted by the Secretary.

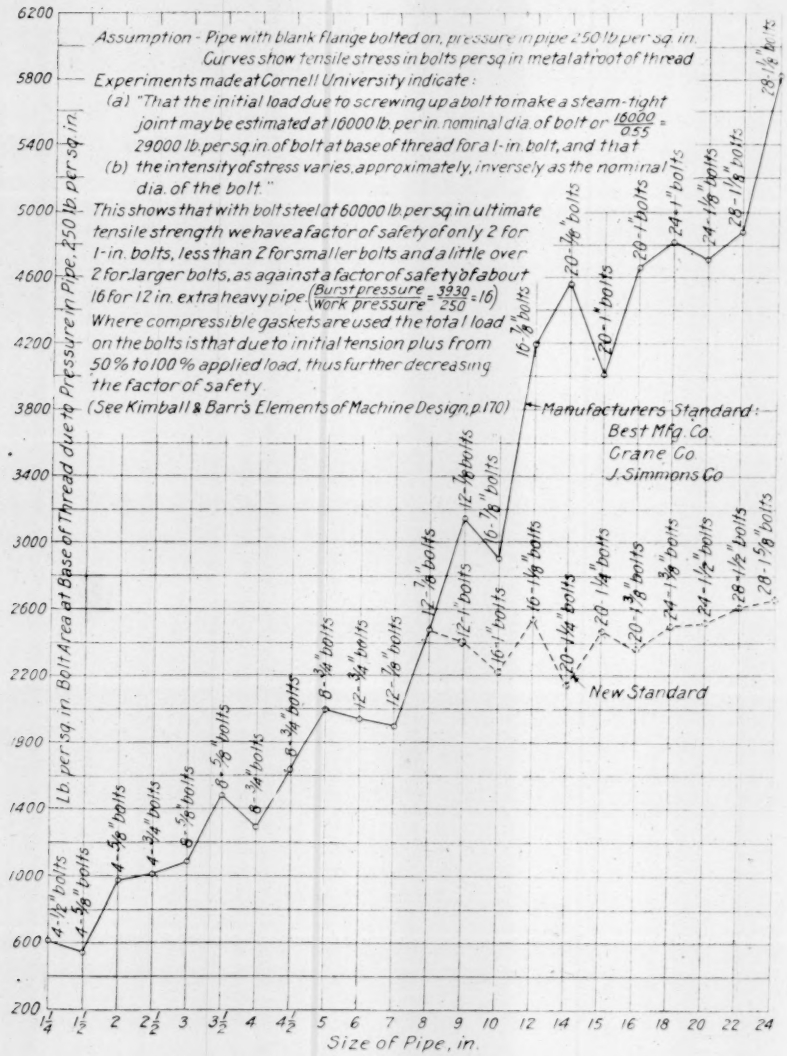
CALVIN W. RICE, *Secretary*

REGARDING THE REPORT ON FLANGES

In connection with the report of the Committee on Flanges, published in this number, the accompanying chart is given showing stresses in the bolts at the base of the thread due to a pressure in the pipe of 250 lb. per sq. in. The chart was plotted by the committee in order to indicate graphically the extent to which these stresses increase with the sizes of the pipe in the Manufacturers' Standard, and the dotted lines show the departure made from the latter in the New Standard in order to keep the stresses down to what is considered by them to be a safe limit, and shows clearly the reason for the radical changes made in the larger sizes.

At the annual meeting of the American Society of Heating and

Ventilating Engineers held last month, a committee examined the new schedule of standard weight and extra heavy flanged fittings


Assumption - Pipe with blank flange bolted on, pressure in pipe 250 lb per sq in.
Curves show tensile stress in bolts per sq in metal at root of thread
Experiments made at Cornell University indicate:
(a) "That the initial load due to screwing up a bolt to make a steam-tight joint may be estimated at 16000 lb. per in. nominal dia. of bolt or 16000/0.55 = 29000 lb. per sq in. of bolt at base of thread for a 1-in. bolt, and that
(b) the intensity of stress varies, approximately, inversely as the nominal dia. of the bolt."
This shows that with bolt steel at 60000 lb. per sq in ultimate tensile strength we have a factor of safety of only 2 for 1-in. bolts, less than 2 for smaller bolts and a little over 2 for larger bolts, as against a factor of safety of about 16 for 12 in. extra heavy pipe (Burst pressure / Work pressure = 3930/250 = 16)
Where compressible gaskets are used the total load on the bolts is that due to initial tension plus from 50% to 100% applied load, thus further decreasing the factor of safety.
(See Kimball & Barr's Elements of Machine Design, p. 170)
Manufacturers Standard:
Best Mfg. Co.
Crane Co.
J. Simmons Co.
New Standard
Lb. per sq in. Bolt Area at Base of Thread due to Pressure in Pipe, 250 lb. per sq in.
Size of Pipe, in.

and flanges, and have recommended to the members of their society that they use it in their designs.

SCHEDULE OF STANDARD WEIGHT AND EXTRA HEAVY FLANGED FITTINGS

THE 1912 U. S. STANDARD

ADOPTED OCTOBER 25, 1911, BY A COMMITTEE OF THE NATIONAL ASSOCIATION OF MASTER STEAM AND HOT WATER FITTERS AND OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

EXPLANATORY NOTES

1 Standard or extra heavy reducing elbows carry same dimensions center to face as regular elbows of largest straight size.

2 Standard or extra heavy tees, crosses and laterals, reducing on run, carry same dimensions face to face as largest straight size.

3 If flanged fittings for lower working pressures than 125 lb. are made, they shall conform in all dimensions, except thickness of shell, to this standard, and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be standard dimensions.

4 Where long-turn fittings are specified, it has reference only to elbows, which are made in two center to face dimensions, to be known as "elbows" and "long-turn" elbows, the latter being used only when so specified.

5 All standard weight fittings must be guaranteed for 125 lb. and extra heavy fittings for 250 lb. working pressure, and each fitting must have some mark cast on it indicating the maker and guaranteed working steam pressure.

6 All extra heavy fittings and flanges to have a raised surface $\frac{1}{16}$ in. high inside of bolt holes for gasket.

Standard weight fittings and flanges to be plain faced.

Bolts to be $\frac{1}{8}$ in. smaller in diameter than bolt holes.

Bolt holes should straddle center lines.

7 Size of all fittings scheduled indicates inside diameter of ports. For outside diameter pipe use corresponding size of inside diameter fittings.

8 The face to face dimension of a reducer, either straight or eccentric, shall be equal to the diameter of the larger flange.

9 Square head bolts with hexagonal nuts are recommended.

10 Twin ells, double branch ells, side outlet ells, side outlet tees and four-way tees, whether straight sizes or reducing, carry same dimensions center to face and face to face as regular ells and tees.

11 Bull head tees or tees increasing on outlet will have same center to face and face to face dimensions as a straight fitting of the size of the outlet.

12 Up to and including the 4 in. size, center to face and face to face dimensions

of reducing fittings will be the same as that of a straight fitting of the larger opening. (See table for reducing fittings larger than 4 in.)

STEEL FLANGES, FITTINGS AND VALVES ARE RECOMMENDED FOR SUPERHEATED STEAM.

Committees on Standardization

William J. Baldwin, Jr.	*National Association of Master Steam and Hot Water Fitters*
N. Loring Danforth	
William T. English	
William P. Kirk	
Thomas B. Cryer	
Edward B. Denny	
William H. Oakes	
H. G. Stott	*The American Society of Mechanical Engineers*
John H. Sparrow	
Wm Schwanhausser	
Walter M. McFarland	
Albert C. Ashton	

SCHEDULE OF STANDARD WEIGHT FLANGED FITTINGS

ADOPTED OCTOBER 25, 1911, BY A COMMITTEE OF THE

NATIONAL ASSOCIATION OF MASTER STEAM AND HOT WATER FITTERS

AND OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
1 inch					
Center to Face	3½	2	5½	3½	2 & 6*
Face to Face	....	....	....	7	8
Diameter of Flange	4	4	4	4	4
Thickness of Flange	7/16	7/16	7/16	7/16	7/16
Diameter of Bolt Circle	3	3	3	3	3
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	9/16	9/16	9/16	9/16	9/16
1¼ inch					
Center to Face	3¾	2	6	3¾	2¼&6¼*
Face to Face	....	....	....	7½	8½
Diameter of Flange	4½	4½	4½	4½	4½
Thickness of Flange	½	½	½	½	½
Diameter of Bolt Circle	3⅜	3⅜	3⅜	3⅜	3⅜
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	9/16	9/16	9/16	9/16	9/16
1½ inch					
Center to Face	4	2¼	6½	4	2¾&6⅞*
Face to Face	....	....	....	8	9¼
Diameter of Flange	5	5	5	5	5
Thickness of Flange	9/16	9/16	9/16	9/16	9/16
Diameter of Bolt Circle	3⅞	3⅞	3⅞	3⅞	3⅞
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	⅝	⅝	⅝	⅝	⅝
2 inch					
Center to Face	4½	2½	7	4½	2½ & 8*
Face to Face	....	....	....	9	10½
Diameter of Flange	6	6	6	6	6
Thickness of Flange	⅝	⅝	⅝	⅝	⅝
Diameter of Bolt Circle	4¾	4¾	4¾	4¾	4¾
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	¾	¾	¾	¾	¾

Asterisk (*) indicates center to face dimensions of run on laterals or double laterals, each way. The smaller dimension is also the center to face dimension of a Y on the short end. All other dimensions on Y's are the same as for elbows of the same size. A reducing Y carries same dimensions as a straight size Y of the size of the largest opening in the fitting in question.

STANDARD WEIGHT FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
2½ inch					
Center to Face	5	2¾	7½	5	2½&9½*
Face to Face	....	....	....	10	12
Diameter of Flange	7	7	7	7	7
Thickness of Flange	11/16	11/16	11/16	11/16	11/16
Diameter of Bolt Circle	5½	5½	5½	5½	5½
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	¾	¾	¾	¾	¾
3 inch					
Center to Face	5½	3	8	5½	3& 10*
Face to Face	....	....	....	11	13
Diameter of Flange	7½	7½	7½	7½	7½
Thickness of Flange	¾	¾	¾	¾	¾
Diameter of Bolt Circle	6	6	6	6	6
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	¾	¾	¾	¾	¾
3½ inch					
Center to Face	6	3½	9	6	3&11½*
Face to Face	....	....	....	12	14½
Diameter of Flange	8½	8½	8½	8½	8½
Thickness of Flange	15/16	15/16	15/16	15/16	15/16
Diameter of Bolt Circle	7	7	7	7	7
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	¾	¾	¾	¾	¾
4 inch					
Center to Face	6½	4	10	6½	3& 12*
Face to Face	....	....	....	13	15
Diameter of Flange	9	9	9	9	9
Thickness of Flange	15/16	15/16	15/16	15/16	15/16
Diameter of Bolt Circle	7½	7½	7½	7½	7½
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
4½ inch					
Center to Face	7	4¼	11	7	3&12½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	11⅞
Center to Face of Outlet, outlets 4½ in.	....	....	....	....	12½
Face to Face	....	....	....	14	15½
Diameter of Flange	9¼	9¼	9¼	9¼	9¼
Thickness of Flange	15/16	15/16	15/16	15/16	15/16
Diameter of Bolt Circle	7¾	7¾	7¾	7¾	7¾
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
5 inch					
Center to Face	7½	4½	12	7½	3½&13½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	12¼
Center to Face of Outlet, outlets 4½ in. and 5 in.	....	....	....	....	13½
Face to Face	....	....	....	15	17
Diameter of Flange	10	10	10	10	10
Thickness of Flange	15/16	15/16	15/16	15/16	15/16
Diameter of Bolt Circle	8½	8½	8½	8½	8½
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞

STANDARD WEIGHT FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
6 inch					
Center to Face	8	5	13	8	3½&14½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	13
Center to Face of Outlet, outlets 4½ in. to 6 in.	....	....	....	....	14½
Face to Face	....	....	....	16	18
Diameter of Flange	11	11	11	11	11
Thickness of Flange	1	1	1	1	1
Diameter of Bolt Circle	9½	9½	9½	9½	9½
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
7 inch					
Center to Face	8½	5½	14½	8½	4&16½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	13¾
Center to Face of Outlet, outlets 4½ in. to 7 in.	...	...	....	....	16½
Face to Face	....	....	....	17	20½
Diameter of Flange	12½	12½	12½	12½	12½
Thickness of Flange	1⅛	1⅛	1⅛	1⅛	1⅛
Diameter of Bolt Circle	10¾	10¾	10¾	10¾	10¾
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
8 inch					
Center to Face	9	6	16	9	4½&17½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	14⅜
Center to Face of Outlet, outlets 4½ in. to 8 in.	....	....	....	....	17½
Face to Face	....	....	....	18	22
Diameter of Flange	13½	13½	13½	13½	13½
Thickness of Flange	1⅛	1⅛	1⅛	1⅛	1⅛
Diameter of Bolt Circle	11¾	11¾	11¾	11¾	11¾
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
9 inch					
Center to Face	10	6½	18	10	4½&19½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	9	17⅝
Center to Face of Outlet, outlets 9 in.	....	....	....	10	19½
Face to Face	....	....	....	20	24
Diameter of Flange	15	15	15	15	15
Thickness of Flange	1⅛	1⅛	1⅛	1⅛	1⅛
Diameter of Bolt Circle	13¼	13¼	13¼	13¼	13¼
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞

STANDARD WEIGHT FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
10 inch					
Center to Face	11	7	20	11	5&20½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	10	18⅝
Center to Face of Outlet, outlets 9 in. and 10 in.	....	....	....	11	20½
Face to Face	....	....	....	22	25½
Diameter of Flange	16	16	16	16	16
Thickness of Flange	1 5/16	1 5/16	1 5/16	1 5/16	1 5/16
Diameter of Bolt Circle	14¼	14¼	14¼	14¼	14¼
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	1	1	1	1	1
12 inch					
Center to Face	12	7½	22	12	5½&24½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	11½	20
Center to Face of Outlet, outlets 9 in. to 12 in.	....	....	....	12	21½
Face to Face	....	....	....	24	30
Diameter of Flange	19	19	19	19	19
Thickness of Flange	1¼	1¼	1¼	1¼	1¼
Diameter of Bolt Circle	17	17	17	17	17
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	1	1	1	1	1
14 inch					
Center to Face	14	7½	24	14	6&27*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	12½	21½
Center to Face of Outlet, outlets 9 in. to 14 in.	....	....	....	14	27
Face to Face	....	....	....	28	33
Diameter of Flange	21	21	21	21	21
Thickness of Flange	1⅜	1⅜	1⅜	1⅜	1⅜
Diameter of Bolt Circle	18¾	18¾	18¾	18¾	18¾
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	1⅛	1⅛	1⅛	1⅛	1⅛
15 inch					
Center to Face	14½	8	26	14½	6&28½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	13	22⅛
Center to Face of Outlet, outlets 9 in. to 15 in.	....	....	....	14½	28½
Face to Face	....	....	....	29	34½
Diameter of Flange	22¼	22¼	22¼	22¼	22¼
Thickness of Flange	1⅜	1⅜	1⅜	1⅜	1⅜
Diameter of Bolt Circle	20	20	20	20	20
Number of Bolt Holes	16	16	16	16	16
Diameter of Bolt Holes	1⅛	1⅛	1⅛	1⅛	1⅛

STANDARD WEIGHT FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
16 inch					
Center to Face	15	8	28	15	6½&30*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	13½	22⅞
Center to Face of Outlet, outlets 9 in. to 16 in.	....	....	....	15	30
Face to Face	....	....	....	30	36½
Diameter of Flange	23½	23½	23½	23½	23½
Thickness of Flange	1 7/16	1 7/16	1 7/16	1 7/16	1 7/16
Diameter of Bolt Circle	21¼	21¼	21¼	21¼	21¼
Number of Bolt Holes	16	16	16	16	16
Diameter of Bolt Holes	1⅛	1⅛	1⅛	1⅛	1⅛
18 inch					
Center to Face	16½	8½	30	16½	7&32*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	14½	24¼
Center to Face of Outlet, outlets 9 in. to 18 in.	....	....	....	16½	32
Face to Face	....	....	....	33	39
Diameter of Flange	25	25	25	25	25
Thickness of Flange	1 9/16	1 9/16	1 9/16	1 9/16	1 9/16
Diameter of Bolt Circle	22¾	22¾	22¾	22¾	22¾
Number of Bolt Holes	16	16	16	16	16
Diameter of Bolt Holes	1¼	1¼	1¼	1¼	1¼
20 inch					
Center to Face	18	9½	32	18	8&35*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	15½	25¾
Center to Face of Outlet, outlets 9 in. to 20 in.	....	....	....	18	35
Face to Face	....	....	....	36	43
Diameter of Flange	27½	27½	27½	27½	27½
Thickness of Flange	1 11/16	1 11/16	1 11/16	1 11/16	1 11/16
Diameter of Bolt Circle	25	25	25	25	25
Number of Bolt Holes	20	20	20	20	20
Diameter of Bolt Holes	1¼	1¼	1¼	1¼	1¼
22 inch					
Center to Face	20	10	34	20	8½&37½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	16½	27⅛
Center to Face of Outlet, outlets 9 in. to 22 in.	....	....	....	20	37½
Face to Face	....	....	....	40	46
Diameter of Flange	29½	29½	29½	29½	29½
Thickness of Flange	1 13/16	1 13/16	1 13/16	1 13/16	1 13/16
Diameter of Bolt Circle	27¼	27¼	27¼	27¼	27¼
Number of Bolt Holes	20	20	20	20	20
Diameter of Bolt Holes	1¼	1¼	1¼	1¼	1¼

STANDARD WEIGHT FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
24 inch					
Center to Face	22	11	36	22	9&40½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	17½	28½
Center to Face of Outlet, outlets 9 in. to 24 in.	....	....	....	22	40½
Face to Face	....	....	....	44	49½
Diameter of Flange	32	32	32	32	32
Thickness of Flange	1⅞	1⅞	1⅞	1⅞	1⅞
Diameter of Bolt Circle	29½	29½	29½	29½	29½
Number of Bolt Holes	20	20	20	20	20
Diameter of Bolt Holes	1¼	1¼	1¼	1¼	1¼
26 inch					
Center to Face	23	13	39	23	9½&43½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	19½	30¾
Center to Face of Outlet, outlets 9 in. to 26 in.	....	....	....	23	43½
Face to Face	....	....	....	46	53
Diameter of Flange	34¼	34¼	34¼	34¼	34¼
Thickness of Flange	2	2	2	2	2
Diameter of Bolt Circle	31¾	31¾	31¾	31¾	31¾
Number of Bolt Holes	24	24	24	24	24
Diameter of Bolt Holes	1⅜	1⅜	1⅜	1⅜	1⅜
28 inch					
Center to Face	24	14	42	24	10½&46½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	20½	33
Center to Face of Outlet, outlets 9 in. to 28 in.	....	....	....	24	46½
Face to Face	....	....	....	48	57
Diameter of Flange	36½	36½	36½	36½	36½
Thickness of Flange	2⅛	2⅛	2⅛	2⅛	2⅛
Diameter of Bolt Circle	34	34	34	34	34
Number of Bolt Holes	28	28	28	28	28
Diameter of Bolt Holes	1⅜	1⅜	1⅜	1⅜	1⅜
30 inch					
Center to Face	25	15	45	25	11 & 49½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	21½	35½
Center to Face of Outlet, outlets 9 in. to 30 in	....	....	....	25	49½
Face to Face	....	....	....	50	60½
Diameter of Flange	38¾	38¾	38¾	38¾	38¾
Thickness of Flange	2⅛	2⅛	2⅛	2⅛	2⅛
Diameter of Bolt Circle	36	36	36	36	36
Number of Bolt Holes	28	28	28	28	28
Diameter of Bolt Holes	1½	1½	1½	1½	1½

SCHEDULE OF EXTRA HEAVY FLANGED FITTINGS

ADOPTED OCTOBER 25, 1911, BY A COMMITTEE OF THE
NATIONAL ASSOCIATION OF MASTER STEAM AND HOT WATER FITTERS
AND
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
1 inch					
Center to Face	4¼	2½	5½	4¼	2⅝&7⅛*
Face to Face	....	....	....	8½	9¾
Diameter of Flange	4½	4½	4½	4½	4½
Thickness of Flange	11/16	11/16	11/16	11/16	11/16
Diameter of Bolt Circle	3¼	3¼	3¼	3¼	3¼
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	⅝	⅝	⅝	⅝	⅝
1¼ inch					
Center to Face	4¼	2½	6	4¼	2⅝&7⅛*
Face to Face	....	....	....	8½	9¾
Diameter of Flange	5	5	5	5	5
Thickness of Flange	¾	¾	¾	¾	¾
Diameter of Bolt Circle	3¾	3¾	3¾	3¾	3¾
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	⅝	⅝	⅝	⅝	⅝
1½ inch					
Center to Face	4½	2¾	6½	4½	2½&8½*
Face to Face	....	....	....	9	11
Diameter of Flange	6	6	6	6	6
Thickness of Flange	13/16	13/16	13/16	13/16	13/16
Diameter of Bolt Circle	4½	4½	4½	4½	4½
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	¾	¾	¾	¾	¾
2 inch					
Center to Face	5	3	7	5	2½&9½*
Face to Face	....	....	....	10	12
Diameter of Flange	6½	6½	6½	6½	6½
Thickness of Flange	⅞	⅞	⅞	⅞	⅞
Diameter of Bolt Circle	5	5	5	5	5
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	¾	¾	¾	¾	¾
2½ inch					
Center to Face	5½	3¼	7½	5½	2½&10½*
Face to Face	....	....	....	11	13
Diameter of Flange	7½	7½	7½	7½	7½
Thickness of Flange	1	1	1	1	1
Diameter of Bolt Circle	5⅞	5⅞	5⅞	5⅞	5⅞
Number of Bolt Holes	4	4	4	4	4
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
3 inch					
Center to Face	6	3½	8	6	3&11½*
Face to Face	....	....	....	12	14½
Diameter of Flange	8¼	8¼	8¼	8¼	8¼
Thickness of Flange	1⅛	1⅛	1⅛	1⅛	1⅛
Diameter of Bolt Circle	6⅝	6⅝	6⅝	6⅝	6⅝
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞

Asterisk (*) indicates center to face dimensions of run on laterals or double laterals, each way. The smaller dimension is also the center to face dimension of a Y on the short end. All other dimensions on Y's are the same as for elbows of the same size. A reducing Y carries same dimensions as a straight size Y of the size of the largest opening in the fitting in question.

EXTRA HEAVY FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
3½ inch					
Center to Face	6½	4	9	6½	3&12½*
Face to Face	....	....	....	13	15½
Diameter of Flange	9	9	9	9	9
Thickness of Flange	1 5/16	1 5/16	1 5/16	1 5/16	1 5/16
Diameter of Bolt Circle	7¼	7¼	7¼	7¼	7¼
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
4 inch					
Center to Face	7	4¼	10	7	3&13½*
Face to Face	....	....	....	14	16½
Diameter of Flange	10	10	10	10	10
Thickness of Flange	1¼	1¼	1¼	1¼	1¼
Diameter of Bolt Circle	7⅞	7⅞	7⅞	7⅞	7⅞
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
4½ inch					
Center to Face	7½	4½	11	7½	3½&14½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	13⅝
Center to Face of Outlet, outlets 4½ in.	....	....	....	....	14½
Face to Face	....	....	....	15	18
Diameter of Flange	10½	10½	10½	10½	10½
Thickness of Flange	1 5/16	1 5/16	1 5/16	1 5/16	1 5/16
Diameter of Bolt Circle	8½	8½	8½	8½	8½
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
5 inch					
Center to Face	8	5	12	8	3½&16*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	14
Center to Face of Outlet, outlets 4½ in. and 5 in.	....	....	....	....	16
Face to Face	....	....	....	16	19½
Diameter of Flange	11	11	11	11	11
Thickness of Flange	1⅜	1⅜	1⅜	1⅜	1⅜
Diameter of Bolt Circle	9¼	9¼	9¼	9¼	9¼
Number of Bolt Holes	8	8	8	8	8
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞
6 inch					
Center to Face	9	5½	13	9	4&17½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	14⅝
Center to Face of Outlet, outlets 4½ in. to 6 in.	....	....	....	....	17½
Face to Face	....	....	....	18	21½
Diameter of Flange	12½	12½	12½	12½	12½
Thickness of Flange	1 7/16	1 7/16	1 7/16	1 7/16	1 7/16
Diameter of Bolt Circle	10⅝	10⅝	10⅝	10⅝	10⅝
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	⅞	⅞	⅞	⅞	⅞

EXTRA HEAVY FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
7 inch					
Center to Face	9½	5¾	14½	9½	4½&19*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	15⅜
Center to Face of Outlet, outlets 4½ in. to 7 in.	....	....	....	....	19
Face to Face	....	....	....	19	23½
Diameter of Flange	14	14	14	14	14
Thickness of Flange	1½	1½	1½	1½	1½
Diameter of Bolt Circle	11⅞	11⅞	11⅞	11⅞	11⅞
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	1	1	1	1	1
8 inch					
Center to Face	10½	6	16	10½	5&20½*
Center to Face of Outlet, outlets 4 in. and under	....	....	....	....	16
Center to Face of Outlet, outlets 4½ in. to 8 in.	....	....	....	....	20½
Face to Face	....	....	....	21	25½
Diameter of Flange	15	15	15	15	15
Thickness of Flange	1⅝	1⅝	1⅝	1⅝	1⅝
Diameter of Bolt Circle	13	13	13	13	13
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	1	1	1	1	1
9 inch					
Center to Face	11	6½	18	11	5&22¾*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	10½	20⅞
Center to Face of Outlet, outlets 9 in.	....	....	....	11	22¾
Face to Face	....	....	....	22	27¾
Diameter of Flange	16¾	16¾	16¾	16¾	16¾
Thickness of Flange	1¾	1¾	1¾	1¾	1¾
Diameter of Bolt Circle	14	14	14	14	14
Number of Bolt Holes	12	12	12	12	12
Diameter of Bolt Holes	1⅛	1⅛	1⅛	1⅛	1⅛
10 inch					
Center to Face	12	7	20	12	5½&25¼*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	11	21⅝
Center to Face of Outlet, outlets 9 in. and 10 in.	....	....	....	12	25¼
Face to Face	....	....	....	24	30¾
Diameter of Flange	18¼	18¼	18¼	18¼	18¼
Thickness of Flange	1⅞	1⅞	1⅞	1⅞	1⅞
Diameter of Bolt Circle	15¾	15¾	15¾	15¾	15¾
Number of Bolt Holes	16	16	16	16	16
Diameter of Bolt Holes	1⅛	1⅛	1⅛	1⅛	1⅛

EXTRA HEAVY FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
12 inch					
Center to Face	13½	8	22	13½	6&27¾*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	12	23
Center to Face of Outlet outlets 9 in. to 12 in.	....	....	....	13½	27¾
Face to Face	....	....	....	27	33¾
Diameter of Flange	20¾	20¾	20¾	20¾	20¾
Thickness of Flange	2	2	2	2	2
Diameter of Bolt Circle	17¾	17¾	17¾	17¾	17¾
Number of Bolt Holes	16	16	16	16	16
Diameter of Bolt Holes	1¼	1¼	1¼	1¼	1¼
14 inch					
Center to Face	15	9½	24	15	6½&31½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	13	24⅜
Center to Face of Outlet, outlets 9 in. to 14 in.	....	....	....	15	31¼
Face to Face	....	....	....	30	37¾
Diameter of Flange	23½	23½	23½	23½	23½
Thickness of Flange	2⅛	2⅛	2⅛	2⅛	2⅛
Diameter of Bolt Circle	20¼	20¼	20¼	20¼	20¼
Number of Bolt Holes	20	20	20	20	20
Diameter of Bolt Holes	1⅜	1⅜	1⅜	1⅜	1⅜
15 inch					
Center to Face	15½	10	26	15½	6½&33¼*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	13½	25⅛
Center to Face of Outlet, outlets 9 in. to 15 in.	....	....	....	15½	33¼
Face to Face	....	....	....	31	39¾
Diameter of Flange	25	25	25	25	25
Thickness of Flange	2⅜	2⅜	2⅜	2⅜	2⅜
Diameter of Bolt Circle	21½	21½	21½	21½	21½
Number of Bolt Holes	20	20	20	20	20
Diameter of Bolt Holes	1⅜	1⅜	1⅜	1⅜	1⅜
16 inch					
Center to Face	16½	10½	28	16½	7½&34½*
Center to Face of Outlet, outlets 8 in. and under	....	....	....	14	25¾
Center to Face of Outlet, outlets 9 in. to 16 in.	....	....	....	16½	34½
Face to Face	....	....	....	33	42
Diameter of Flange	26	26	26	26	26
Thickness of Flange	2¼	2¼	2¼	2¼	2¼
Diameter of Bolt Circle	22½	22½	22½	22½	22½
Number of Bolt Holes	20	20	20	20	20
Diameter of Bolt Holes	1½	1½	1½	1½	1½

EXTRA HEAVY FLANGED FITTINGS—Continued

	Elbows	45-Deg. Elbows	Long Turn Elbows	Tees and Crosses	Single and Double Laterals or Y Branches, and Y's
18 inch					
Center to Face	17¾	11	30	17¾	8&37¾*
Center to Face of Outlet, outlets 8 in. and under				15	27⅛
Center to Face of Outlet outlets 9 in. to 18 in.				17¾	37¾
Face to Face				35½	45¾
Diameter of Flange	28½	28½	28½	28½	28½
Thickness of Flange	2⅜	2⅜	2⅜	2⅜	2⅜
Diameter of Bolt Circle	24¾	24¾	24¾	24¾	24¾
Number of Bolt Holes	24	24	24	24	24
Diameter of Bolt Holes	1½	1½	1½	1½	1½
20 inch					
Center to Face	19¼	11½	32	19¼	8½&40¾*
Center to Face of Outlet, outlets 8 in. and under				16	28⅝
Center to Face of Outlet, outlets 9 in. to 20 in.				19¼	40¾
Face to Face				38½	49¼
Diameter of Flange	31	31	31	31	31
Thickness of Flange	2½	2½	2½	2½	2½
Diameter of Bolt Circle	27	27	27	27	27
Number of Bolt Holes	24	24	24	24	24
Diameter of Bolt Holes	1⅝	1⅝	1⅝	1⅝	1⅝
22 inch					
Center to Face	20½	12	34	20½	9½&43¾*
Center to Face of Outlet, outlets 8 in. and under				17	30
Center to Face of Outlet outlets 9 in. to 22 in.				20½	43¾
Face to Face				41	53¼
Diameter of Flange	33	33	33	33	33
Thickness of Flange	2⅝	2⅝	2⅝	2⅝	2⅝
Diameter of Bolt Circle	29¼	29¼	29¼	29¼	29¼
Number of Bolt Holes	28	28	28	28	28
Diameter of Bolt Holes	1⅝	1⅝	1⅝	1⅝	1⅝
24 inch					
Center to Face	22½	13	36	22½	10&47½*
Center to Face of Outlet, outlets 8 in. and under				18	31½
Center to Face of Outlet, outlets 9 in. to 24 in.				22½	47½
Face to Face				45	57½
Diameter of Flange	36	36	36	36	36
Thickness of Flange	2¾	2¾	2¾	2¾	2¾
Diameter of Bolt Circle	32	32	32	32	32
Number of Bolt Holes	28	28	28	28	28
Diameter of Bolt Holes	1¾	1¾	1¾	1¾	1¾

SCHEDULE OF STANDARD FLANGES

FOR STEAM PRESSURES UP TO 125 LB. PER SQ. IN.

ADOPTED OCTOBER 25, 1911, BY A COMMITTEE OF THE

NATIONAL ASSOCIATION OF MASTER STEAM AND HOT WATER FITTERS

AND OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

All dimensions are in inches

Size of Pipe	Diameter of Flange	Thickness of Flange	Diameter of Bolt Circle	Number of Bolts	Size of Bolts	Diameter of Bolt Holes
1	4	7/16	3	4	7/16	9/16
1 1/4	4 1/2	1/2	3 3/8	4	7/16	9/16
1 1/2	5	9/16	3 7/8	4	1/2	5/8
2	6	5/8	4 3/4	4	5/8	3/4
2 1/2	7	11/16	5 1/2	4	5/8	3/4
3	7 1/2	3/4	6	4	5/8	3/4
3 1/2	8 1/2	13/16	7	4	5/8	3/4
4	9	15/16	7 1/2	8	5/8	3/4
4 1/2	9 1/4	15/16	7 3/4	8	5/8	3/4
5	10	15/16	8 1/2	8	3/4	7/8
6	11	1	9 1/2	8	3/4	7/8
7	12 1/2	1 1/16	10 3/4	8	3/4	7/8
8	13 1/2	1 1/8	11 3/4	8	3/4	7/8
9	15	1 1/8	13 1/4	12	3/4	7/8
10	16	1 3/16	14 1/4	12	7/8	1
12	19	1 1/4	17	12	7/8	1
14 O.D.	21	1 3/8	18 3/4	12	1	1 1/8
15 O.D.	22 1/4	1 3/8	20	16	1	1 1/8
16 O.D.	23 1/2	1 7/16	21 1/4	16	1	1 1/8
18 O.D.	25	1 9/16	22 3/4	16	1 1/8	1 1/4
20 O.D.	27 1/2	1 11/16	25	20	1 1/8	1 1/4
22 O.D.	29 1/2	1 13/16	27 1/4	20	1 1/8	1 1/4
24 O.D.	32	1 7/8	29 1/2	20	1 1/8	1 1/4
26 O.D.	34 1/4	2	31 3/4	24	1 1/4	1 3/8
28 O.D.	36 1/2	2 1/16	34	28	1 1/4	1 3/8
30 O.D.	38 3/4	2 1/8	36	28	1 3/8	1 1/2

Bolt holes should straddle center lines.

Flanges should be plain faced.

SCHEDULE OF EXTRA HEAVY FLANGES FOR EXTRA HEAVY FITTINGS AND VALVES

FOR STEAM PRESSURES FROM 125 TO 250 LB. PER SQ. IN.

ADOPTED OCTOBER 25 1911, BY A COMMITTEE OF THE

NATIONAL ASSOCIATION OF MASTER STEAM AND HOT WATER FITTERS

AND OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

All dimensions are in inches

Size of Pipe	Diameter of Flange	Thickness of Flange	Diameter of Bolt Circle	Number of Bolts	Size of Bolts	Diameter of Bolt Holes
1	4 1/2	1/16	3 1/4	4	1/2	5/8
1 1/4	5	3/4	3 3/4	4	1/2	5/8
1 1/2	6	5/16	4 1/2	4	5/8	3/4
2	6 1/2	7/8	5	4	5/8	3/4
2 1/2	7 1/2	1	5 7/8	4	3/4	7/8
3	8 1/4	1 1/8	6 5/8	8	3/4	7/8
3 1/2	9	1 3/16	7 1/4	8	3/4	7/8
4	10	1 1/4	7 7/8	8	3/4	7/8
4 1/2	10 1/2	1 5/16	8 1/2	8	3/4	7/8
5	11	1 3/8	9 1/4	8	3/4	7/8
6	12 1/2	1 7/16	10 5/8	12	3/4	7/8
7	14	1 1/2	11 7/8	12	7/8	1
8	15	1 5/8	13	12	7/8	1
9	16 3/4	1 3/4	14	12	1	1 1/8
10	18 1/4	1 7/8	15 3/4	16	1	1 1/8
12	20 3/4	2	17 3/4	16	1 1/8	1 1/4
14 O.D.	23 1/2	2 1/8	20 1/4	20	1 1/4	1 3/8
15 O.D.	25	2 3/16	21 1/2	20	1 1/4	1 3/8
16 O.D.	26	2 1/4	22 1/2	20	1 3/8	1 1/2
18 O.D.	28 1/2	2 3/8	24 3/4	24	1 3/8	1 1/2
20 O.D.	31	2 1/2	27	24	1 1/2	1 5/8
22 O.D.	33	2 5/8	29 1/4	28	1 1/2	1 5/8
24 O.D.	36	2 3/4	32	28	1 5/8	1 3/4

Bolt Holes should straddle center lines.

Flanges should have 1/16 inch raised face for gaskets.

Square Head Bolts with hexagonal nuts are recommended.

STUDENT BRANCHES

ARMOUR INSTITUTE OF TECHNOLOGY

The semi-annual banquet of the Armour Institute Student Branch was held on December 20, and was attended by the faculty of the Mechanical Department and by the Junior and Senior Students.

At a professional session on January 11, A. W. Semerak gave a talk on Steel Belting, illustrated by lantern slides. He stated that steel belts have been developed to the greatest extent in Germany where they are used for many different kinds of drives, to replace leather belts and rope drives. Examples of successful installations of steel belts were quoted and many details of construction described.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

At a meeting of the Mechanical Engineering Society of the Massachusetts Institute of Technology, on December 19, J. F. Hoxie, Mem. Am. Soc. M. E., spoke on the Cause of the Growth of Fungii in Wood, showing by the aid of numerous slides the different kinds of infected wood and of fungii at different stages of development, and why they grow more rapidly in one piece of wood than in another.

On December 21, B. R. T. Collins, Mem. Am. Soc. M. E., gave a talk on The Economizing of Power Plants. He spoke particularly of his own experience, stating the cause of complaint and his method of examination and procedure, with the results obtained.

OHIO STATE UNIVERSITY

G. H. Mueller, chief engineer of the Jeffrey Manufacturing Company, addressed a meeting of the Ohio State University Student Branch on December 18, upon the subject of Power Plant Machinery, dealing especially with the history of improvements in this field and with the various types of conveyors used. He predicted a great future for concrete in connection with power plant construction. The discussion was participated in by E. A. Hitchcock, F. E. Sanborn, H. Seitz, H. Belt, and W. Assel.

The branch was visited on January 4 by Calvin W. Rice, Secretary of the Society, who met with a cordial reception at the hands of the

students. He spoke principally of the advantages to the students of affiliation with the Society.

POLYTECHNIC INSTITUTE OF BROOKLYN

At a meeting of the branch on December 9, Prof. Wm. T. Magruder, Mem. Am. Soc. M. E., spoke upon Progress in Engineering during 1911. On January 6 the members were addressed by Wm. T. Donnelly, Mem. Am. Soc. M. E., on Floating Dry Docks.

STEVENS INSTITUTE OF TECHNOLOGY

Three lectures by prominent engineers were delivered before the Stevens Engineering Society in the month of December. These were: The Comparative Economics of Coal and Water Gas, by Arthur H. Elliott, on December 6; Vapors for Heat Engines, by Wm. D. Ennis, Mem. Am. Soc. M. E., on December 14; and The Progress of Mechanical Engineering in 1911, by Wm. T. Magruder, Mem. Am. Soc M. E., on December 19.

On December 9 the Branch inspected the power house of the Hudson & Manhattan Railroad in Jersey City, through the courtesy of its officials.

The branch was the guest of the Polytechnic Institute student branch on January 6, when a lecture by Wm. T. Donnelly, Mem. Am. Soc. M. E., was presented. The question of consolidation of these two branches for their mutual strengthening is under consideration.

On January 9 Roy C. Whitall (Student, 1912) presented a paper on The Automatic Telephone, which was discussed by O. C. Roesen, G. L. Clouser, A. E. Bauhan, and Messrs. Langlotz and Stern.

On January 11 Charles Kirchhoff, Mem. Am. Soc. M. E., President of the American Institute of Mining Engineers, delivered a lecture on Factors in the Development of the Iron Industry.

UNIVERSITY OF CINCINNATI

Calvin W. Rice, Secretary of the Society, addressed the University of Cincinnati Student Branch on the benefits of the membership in the Society on January 4. He called attention to the advantages afforded by a student section and emphasized the importance of joining the Society after graduation. Addresses were also made by E. F. Du Brul, Assoc. Am. Soc. M. E., A. L. De Leeuw, Mem. Am. Soc. M. E., Dean Herman Schneider of the College of Engineering, and F. A. Stanley, of The American Machinist.

UNIVERSITY OF KANSAS

In addition to the regular monthly meetings, the University of Kansas Student Branch held its third annual meeting December 14, 1911, characterized by the elaborate preparations in the way of entertainment and the carefully arranged program which give to this organization its unique distinction. The program was as follows: Address of Welcome, F. O. Marvin; The Action of Reciprocating Pump Valves, C. I. Corp, Jun. Am. Soc. M. E.; The Technical School Engineer, Louis Bendit, Mem. Am. Soc. M. E.; Shop Management, B. W. Benedict. Lunch was then served, after which an opportunity was afforded to inspect the new Santa Fé dynamometer car. This was followed by a demonstration of the automatic screw machine by the National Acme Machine Company, Fowler Shops, and a conclusion of the professional program consisting of papers on Drilling Gas Wells, W. M. Welch; The Manufacture of Iron and Steel, W. A. Whitaker, and an address, The Santa Fé Dynamometer Car, H. B. McFarland, Mem. Am. Soc. M. E. A banquet was held in the evening at which Prof. P. E. Walker, Mem. Am. Soc. M. E., presided as toastmaster.

UNIVERSITY OF KENTUCKY

The University of Kentucky Student Branch was addressed by Calvin W. Rice, Secretary of the Society, on January 5, upon the work of the Society and the advantages which it offers to student branches. He was enthusiastically received by the membership.

UNIVERSITY OF MISSOURI

The Student Branch at the University of Missouri enjoyed a visit from Calvin W. Rice, Secretary of the Society, on January 8, and a large audience of faculty members and engineering students listened to a brief address by him on the work of the Society with particular reference to the student branches.

NECROLOGY

CARYL D. HASKINS

Caryl D. Haskins was born at Waltham, Mass., May 22, 1867, and obtained his early education in the public schools of Philadelphia and Dedham, Pa. In 1879 he went to England, where he attended the Allison Towers and Roslyndale schools, devoted to the preparation of boys for military and engineering work, and later studied under the London University tuition system for a year. In 1887 he entered the employ of Haskins, Davis & Company, specializing in gas-engine work. Six months later he entered the shops of S. Z. Ferranti, where he was connected with the early work at the Old Grosvenor and Deptford stations. Returning to the United States in 1889, he was employed as designing draftsman by the Thomson Electric Welding Company, Lynn, Mass., but was soon transferred to the Thomson-Houston Electric Company as assistant foreman of the meter department. Shortly afterwards he was given charge of the company's outside meter interests. When the General Electric Company was formed, Mr. Haskins was put in charge of the meter interests of the consolidated company. In 1893 he was also given charge of the company's instrument department and in 1900 of the switchboard department.

While Mr. Haskins is best known as a meter expert, he was also an inventor in other lines, including the subjects of steel refining, meters, transformers, switches, circuit-breakers, time relays, etc. Perhaps his most important as well as most ingenious invention was an auto-dirigible torpedo, for which he refused several offers from foreign powers.

Mr. Haskins also found time for literary work. He was the author of a book on Transformers and of a work of fiction entitled, For the Queen in South Africa. He also lectured frequently at army and navy institutes. He had made a special study of electricity as applied to offense, and particularly to defense, in time of war and was considered an authority on this subject, and frequently acted in an advisory capacity to the Government in matters of this nature.

Mr. Haskins was a member of the American Institute of Electrical

Engineers and the National Electric Light Association and was an associate member of the Military Service Institute and the United States Naval Academy. He died November 18, 1911.

WARREN S. JOHNSON

Warren S. Johnson was born in Leicester, Vt., November 6, 1847, and died December 5, 1911. At an early age the family moved to Waukesha, Wis., where he attended school and learned the printer's trade. At seventeen years of age he became foreman in a newspaper office at Durand, Wis., at the same time attending high school. He taught school in Menominee for a number of years and then became county superintendent of schools. In 1876 he was appointed professor of mathematics, sciences and drawing at the state normal school at Whitewater, Wis. Here he maintained a private laboratory where he experimented with storage batteries and where he conceived the idea of automatic temperature regulation. In 1883 he resigned his professorship and formed a partnership with William Plankinton of Milwaukee, where he developed the Johnson system of temperature regulation, now used all over the world. In 1885 the partnership became a corporation known as the Johnson Electric Service Company, the name being changed later to the Johnson Service Company.

The temperature regulator was not Mr. Johnson's only invention. Others were a brass foundry furnace, gasolene gas machine, pneumatic clock and time system, wireless telegraph system, automobile parts and various devices used in connection with the business in which he was engaged. The huge pneumatic clock at the St. Louis Exposition in 1904 was made by Mr. Johnson. He also constructed the city hall clock in Philadelphia, the largest in the world, and the city hall clock in Milwaukee which has the largest bell in the world.

Mr. Johnson was a member of the Franklin Institute, the American Institute of Electrical Engineers and the American Society of Heating and Ventilating Engineers.

EDGAR W. MIX

Edgar W. Mix was born at Columbus, Ohio, February 4, 1867, and in 1888 was graduated from the Ohio State University with the degree of B. S. The two years following were spent in the employ of the Thomson-Houston Electric Company, Lynn, Mass., in general shop work and practice and in the development of the Thomson recording watt meter. In 1890 he took this meter to Paris where it was awarded first prize in the meter contest. The following five

years were devoted to general engineering, designing and installing various electric railway and lighting plants in France and Russia. In 1895 he became chief engineer of the Société des Etablissements Postel-Vinay, Paris, which concern was then controlled and has since been absorbed by the Thomson-Houston Electric Company. In this capacity Mr. Mix played an important part in the development of all the electrical machinery which the company have been building since he joined them, as well as steam turbines and modern extraction machinery for collieries. He left the Thomson-Houston Company in October 1911, to take up the European branch of the General Motors Export Company of America.

Mr. Mix was a member of both the Aero Club of France and the Aero Club of America, and represented both countries in the international balloon race from Zurich on October 5, 1909, in which he was the only American contestant. He died suddenly November 12, 1911.

HUGH S. MORRISON

Hugh S. Morrison was born in Petersburg, Va., July 7, 1877, and died at Saranac Lake, N. Y., November 24, 1911. He was educated at the high schools and for two years was a student at the Virginia Mechanics Institute, Richmond. He obtained his shop experience at the Richmond Iron Works, where he was foreman of the boiler shop, and at the Petersburg Iron Works, Roanoke Rapids, N. C., being in charge of the outside construction work. In July 1898 he entered the drawing-room of the Richmond Locomotive Works and later became assistant to the superintendent in charge of the layout and construction of the new Smith boiler, machine and special shops and track system in the yard. He opened an office of his own in 1901, and was for a short time associated with Mr. Kinderwater. During this period he designed the machinery and layout and the heating and ventilating systems for the Baughman Stationery Company, as well as the layout of the entire building. In 1903 he became a member of the firm of Myers, Finney & Morrison, designing and constructing special machinery, and was responsible for the drawings and specifications for the Keely cotton compress for that company, Chester, S. C. Mr. Morrison organized the Morrison Machinery & Supply Company of Richmond, Va., in 1905, with which he was actively connected until his death.

THOMAS F. SALTER

Thomas F. Salter was born in Prattsville, N. Y., March 31, 1875,

and after his graduation from the Syracuse, N. Y., high school, learned his trade in the machine shops of the Syracuse Specialty & Manufacturing Company, and through correspondence schools and private tutors. The several years following he was employed by the General Electric Company, Schenectady, N. Y., and by the Western Electric Company, New York City. From 1902 to 1906 he was assistant to the executive engineer of the C. W. Hunt Company, West New Brighton, N. Y., being instrumental in designing machinery for a number of coal-hoisting installations for the United States Government, various railroads, and other users of such apparatus, and was considered by them an engineer and designer of unusual ability. He then entered the employ of the North Penn Iron Company, Philadelphia, Pa., as chief engineer and manager of the shop. While here he designed and constructed a special 20-ton hand-traveling crane for the Cambria Steel Company, intended for operation in a building of semicircular form. He was also responsible for three 5-ton 3-motor electric traveling cranes for the Bureau of Supplies and Accounts, Navy Department, which were erected at the Charleston Navy Yard. At the time of his death, October 25, 1911, he was manager of sales and engineering with the Standard Roller Bearing Company, Philadelphia, Pa.

Mr. Salter was a member of the New York Railroad Club, the Society of Automobile Engineers, the Interurban and Street Railway Association and the Sales Managers Association.

DESIGN AND MECHANICAL FEATURES OF THE CALIFORNIA GOLD DREDGE

By Robert E. Cranston

ABSTRACT OF PAPER

Detailed information in regard to all conditions is requisite in order to properly design a gold dredge. The first California dredge, built in 1897–1898, was patterned after the New Zealand type, and the present California type dredge is a combination of this type and several others.

The buckets are made with cast steel base, pressed steel hood and manganese steel lip. The lower tumbler is a 6-sided steel casting with renewable wearing plates over which the buckets pass. They then travel up a structural steel ladder on rollers and over a 6-sided upper tumbler which is driven by a chain of cast-steel gears. The buckets dump their material into a hopper which discharges into a shaking or revolving screen. The fine material goes through onto gold-saving tables and the coarse material is stacked aft of the dredge by means of a belt conveyor. The dredge is held in place by steel spuds and moved by means of side lines running to a motor-driven winch. The digging ladder is raised by means of a separate winch driven by the digging motor. Water is furnished under pressure by motor-driven centrifugal pumps. The hull is built of Douglas fir and should be designed last.

For rough estimating the cost of a modern California type dredge can be figured at about $180 per ton.

THE DESIGN AND MECHANICAL FEATURES OF THE CALIFORNIA GOLD DREDGE

By Robert E. Cranston, Sacramento, Cal.

Member of the Society

This paper deals only with that branch of gold dredging which comes directly within the province of mechanical engineering, and the designs described are those commonly used in California. No attempt is made to cover the whole field, and the author has recorded only his personal observations of the more important mechanical features.

2 In designing a gold dredge the first essentials to be determined are the quantity of material to be handled and the depth to which it will be necessary to dig. To a lesser degree, the size and quantity of boulders, quantity of clay, nature and contour of bed rock, contour of the surface, whether the dredge is to be operated in a river or in an inland pond, how high above the water line it will be necessary to maintain the bank, the nature and consistency of the gravel, distance from a railroad or other cheap transportation, costs of lumber, labor, electric power, supplies, etc., condition of wagon roads, distance and transportation facilities to a first class machine shop, climate, distance to nearest public power line, and possibilities of developing electric power on or near the property, are important. If electric power is not available, then the costs of the various fuels for generating steam or running gas engines must be considered, and if the dredge is to be operated inland, how much water is available, and any other data peculiar to the property.

3 All the above information is requisite for the proper design of a dredge to do the most efficient work. A designer should insist on a full knowledge of all the conditions, otherwise his product is likely to be more or less unsuited to the work it has to do.

4 Beginning with the capacity required, the dimensions of the bucket bases, pins, lips, etc., and the horsepower of the digging

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motor are determined. The depth of ground determines the length of ladder, which is modified by the height to which it is desired to elevate in order to wash and dispose of the gravel properly. The screen pumps and stacker are designed according to the quantity and quality of coarse material and the height to which it is to be stacked. The other machinery is designed to suit the local conditions, and the hull is planned to support the machinery properly, its dimensions varying greatly from the small capacity shallow diggers, to the 16-ft. dredges excavating 60 or 70 ft. below water line.

5 The mechanical equipment of the California gold dredge as it is today has in it much to be desired in the way of correct dimensions and the relative strengths of its component parts. Many of these parts should be made stronger, likewise there is much waste material which could just as well be done away with. Notwithstanding these admitted faults the California gold dredge has during the last few years shown vast improvement over the earlier attempts, and with all its frailties has many good points in its make-up which might be profitably studied by engineers called upon to design similar pieces of machinery.

6 The first successful gold dredges in California were patterned after the New Zealand type. A brief description of one of these, built in 1897 and 1898, will not perhaps be out of place (Fig. 1).

7 The hull was built of wood of a rectangular cross-section, having an opening or well extending along the center line part way back from the bow. Near the after end of this well was built an elevated framed structure or main gantry which served to support one end of the digging ladder, the other end being suspended by wire rope blocks from a head frame or "bow gantry" spanning the well. This ladder was a plate girder upon which were mounted rollers to carry the bucket line. The bucket line was composed of alternate links and buckets pinned together to form an endless chain. The buckets passed over a four-sided flanged tumbler keyed to a shaft and driven by a spur gear mounted on the main gantry. At the lower end of the ladder another tumbler served as an idler. In going over the upper tumbler the buckets dumped into a hopper leading into an inclined revolving screen with $\frac{3}{8}$-in. to $\frac{5}{8}$-in. perforations. A spray pipe ran lengthways through the screen, throwing jets of water upon the gravel and washing the finer particles and gold through the screen holes upon gold-saving tables. These tables were covered with cocoanut matting held in place by expanded metal and were arranged so as first to distribute the fine material over a considerable area and

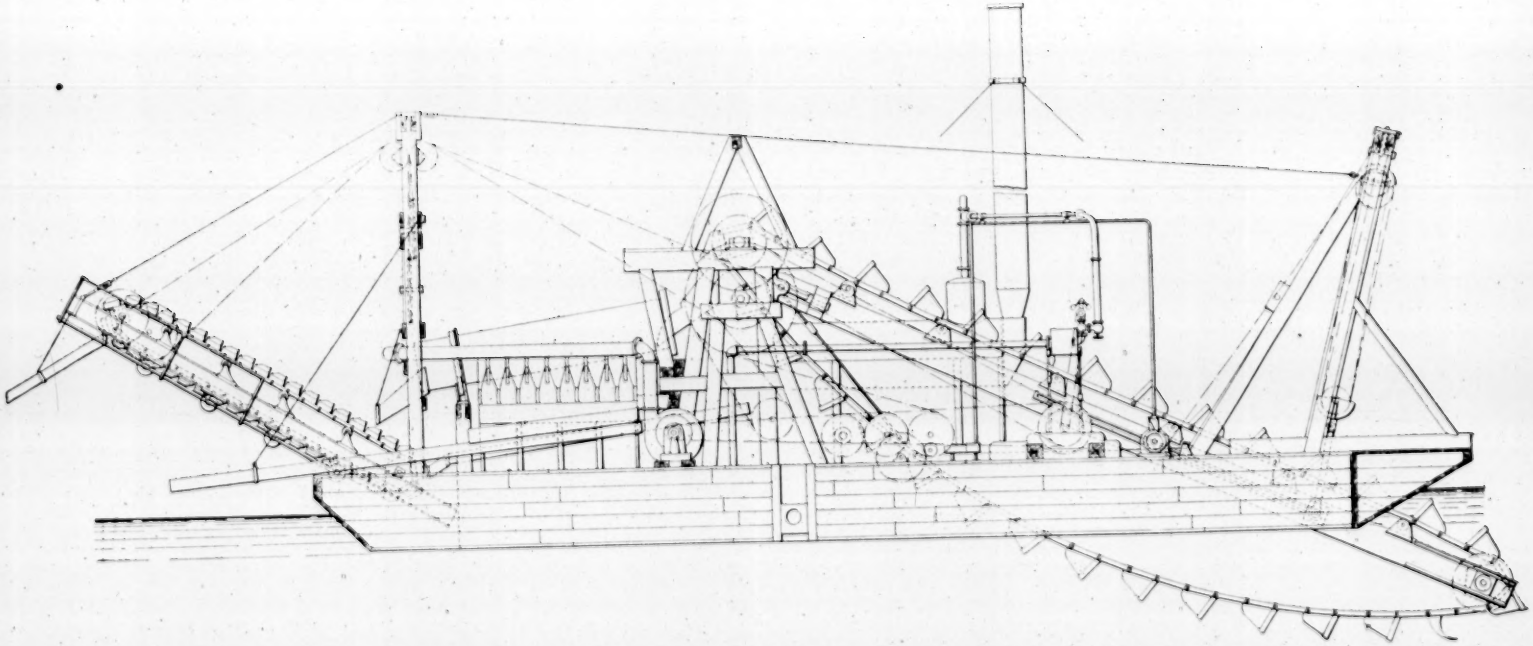


FIG. 1 FIRST SUCCESSFUL CALIFORNIA DREDGE, 1897–1898

then discharge it at the stern; the oversize went to a bucket elevator and was stacked some distance aft of the dredge. The dredge was held up to its work by a head line which could be reeled in by means of a drum on the dredge. Side movement was secured by lines, two at the bow and two at the stern, each running to a separate drum. The motive power for these dredges was steam, which was soon replaced by electricity.

8 Three years before this type of dredge was introduced in California, one of the type known as the double-lift was built near Ban-

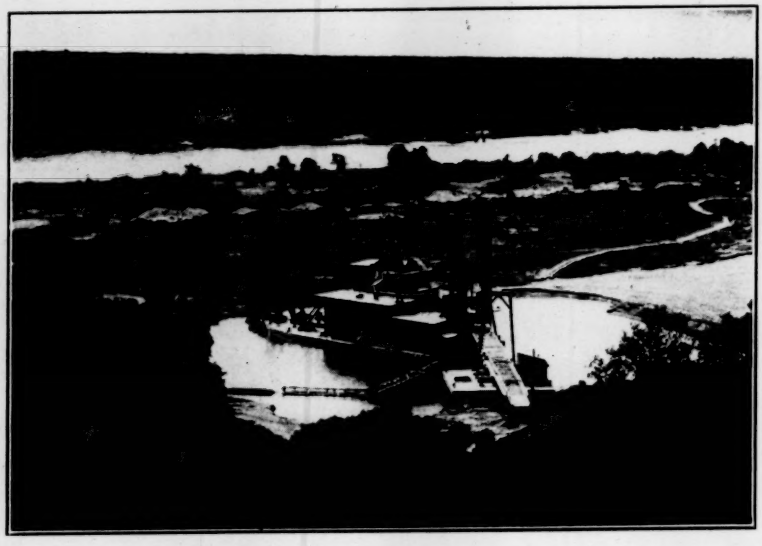
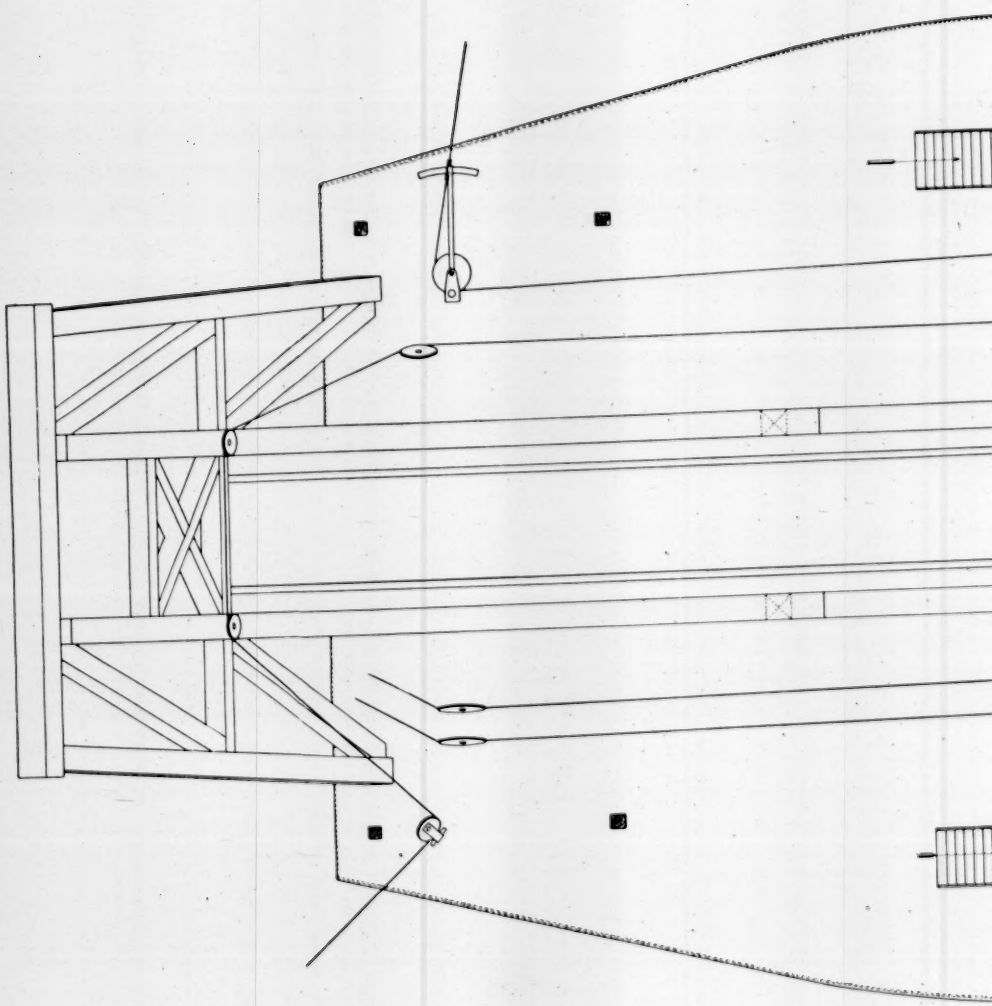


FIG. 2 SINGLE-LIFT CALIFORNIA GOLD DREDGE, 1900

nock, Mont. The main gantry was much lower than in the New Zealand type. A revolving screen was used, but the holes were much larger, so that rocks 4 in. to 6 in. in diameter would pass through. The oversize was dumped at the side of the dredge, and the undersize allowed to collect in a sump from whence it was elevated by means of a centrifugal pump into a sluice running aft from near the throat of the well. At the stern was a pivot connection with another sluice mounted on an auxiliary scow. This scow could be moved from side to side, one end of its sluice being pivoted at the stern of the dredge and the other moving in the arc of a circle, thus distributing the gravel over a considerable area. Instead of a head line, spuds



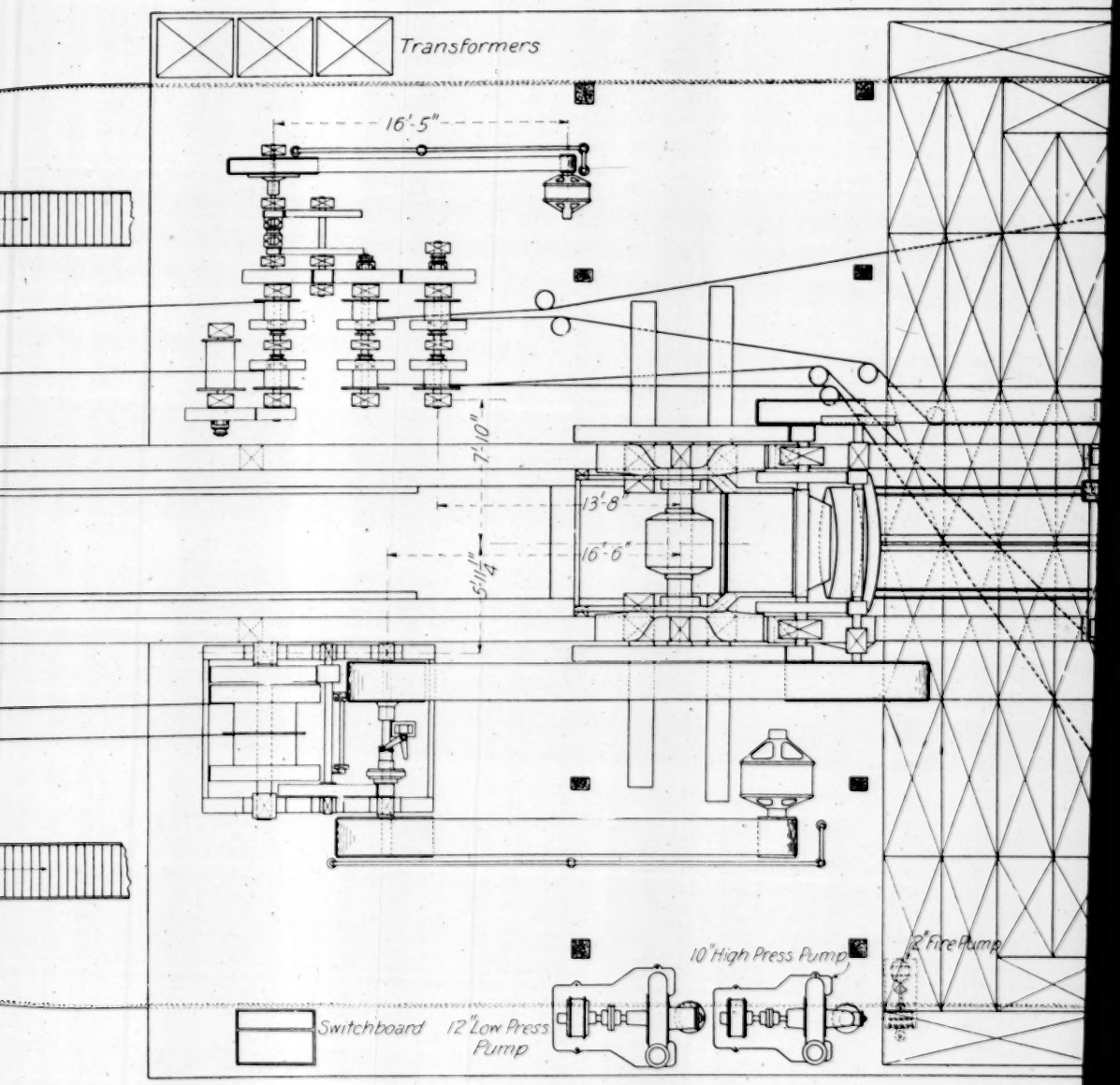


FIG. 4 PLAN, 9-CU. FT. DREDGE

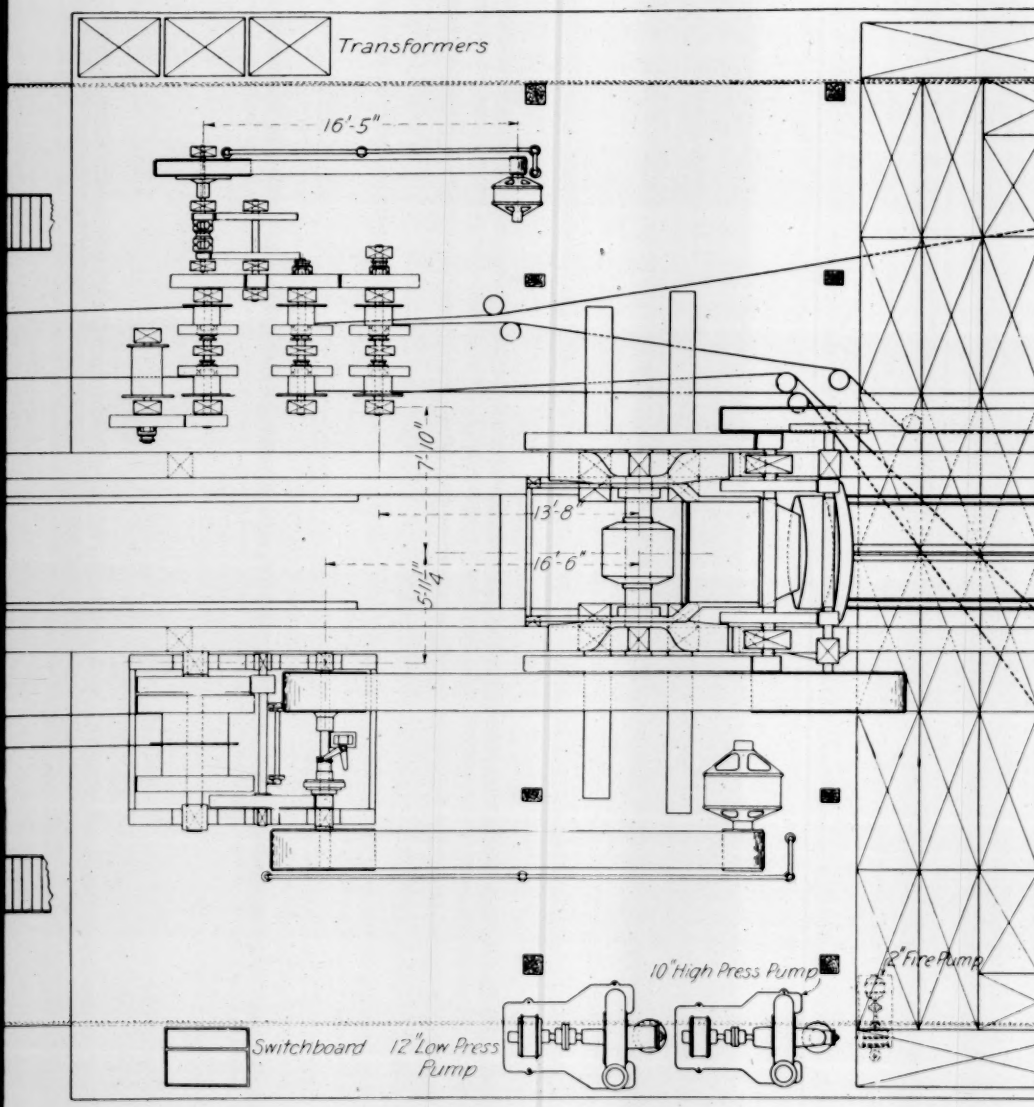
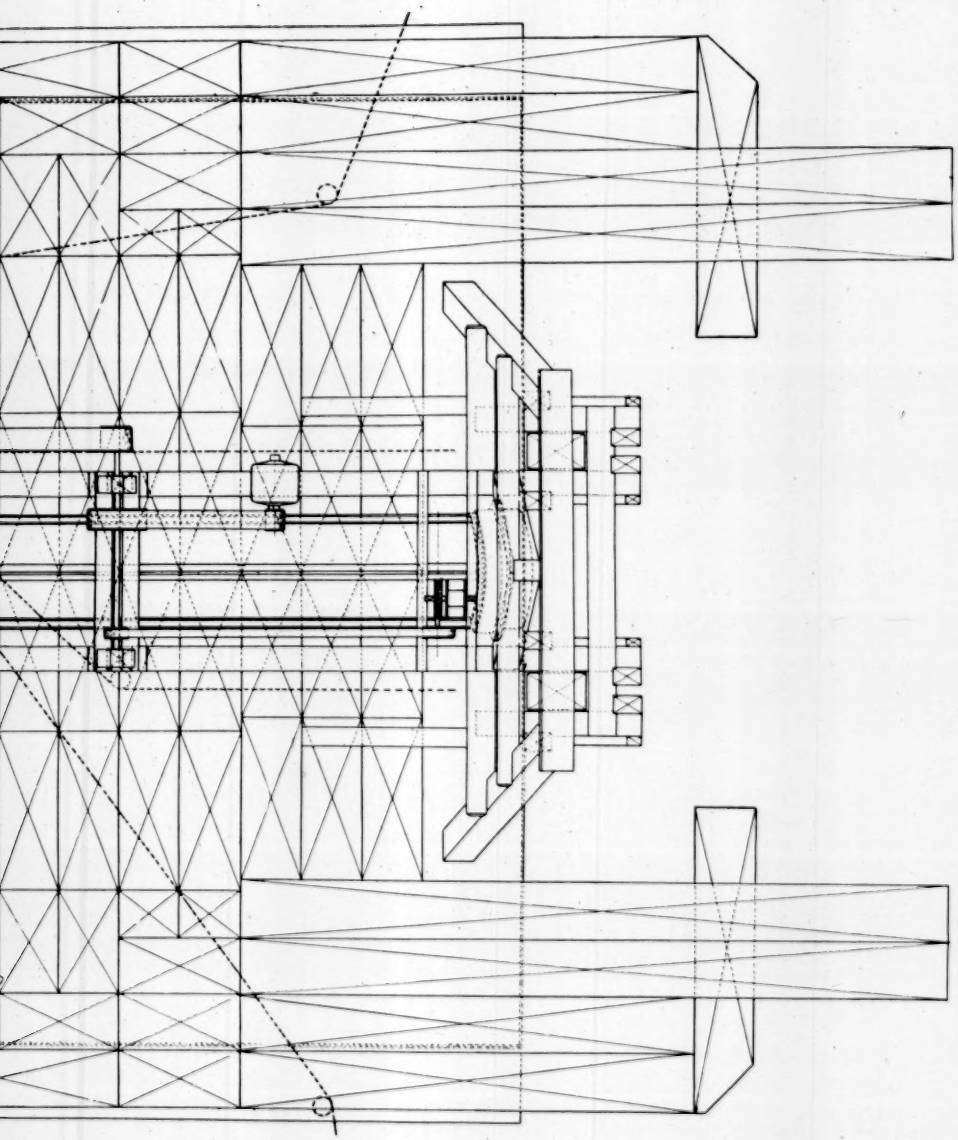
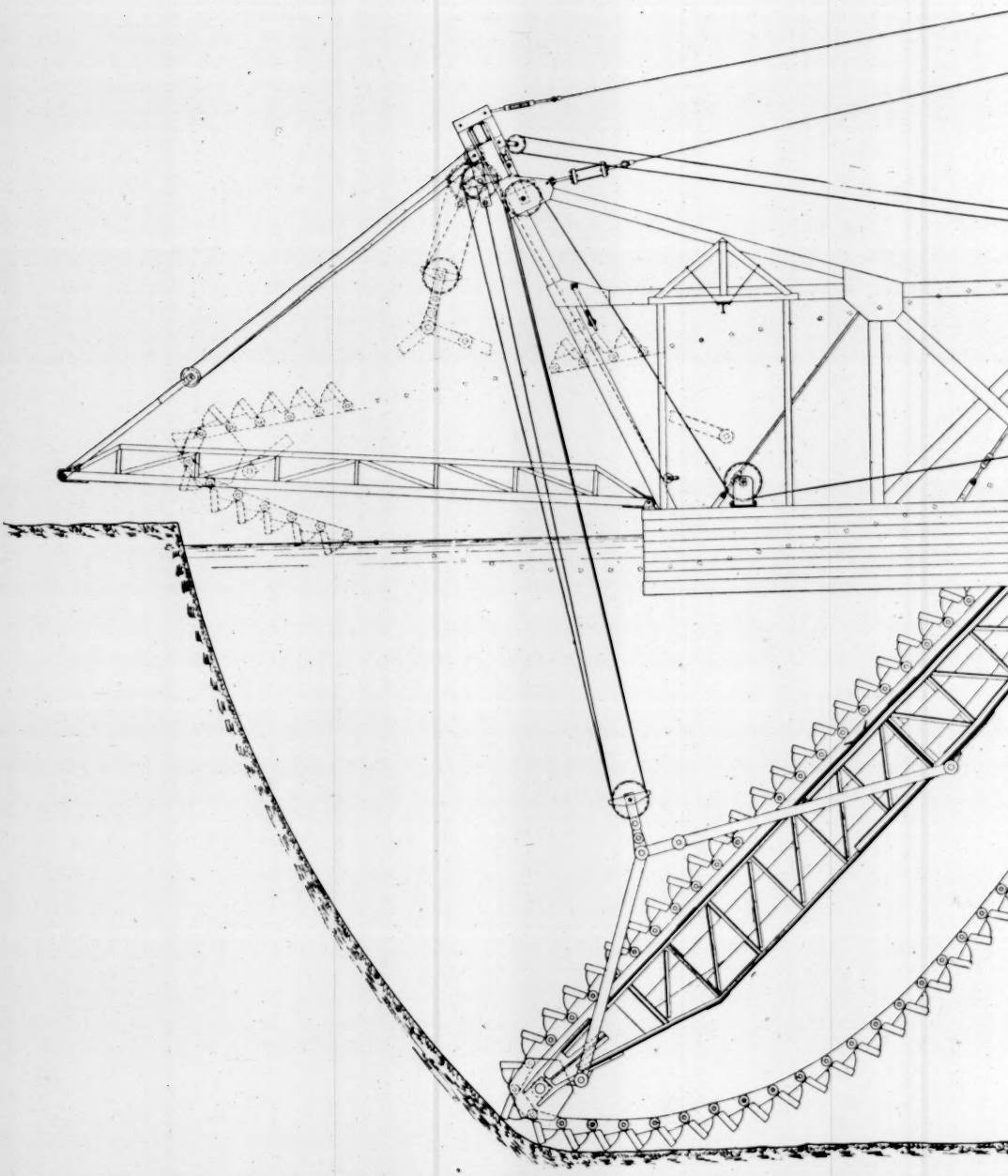
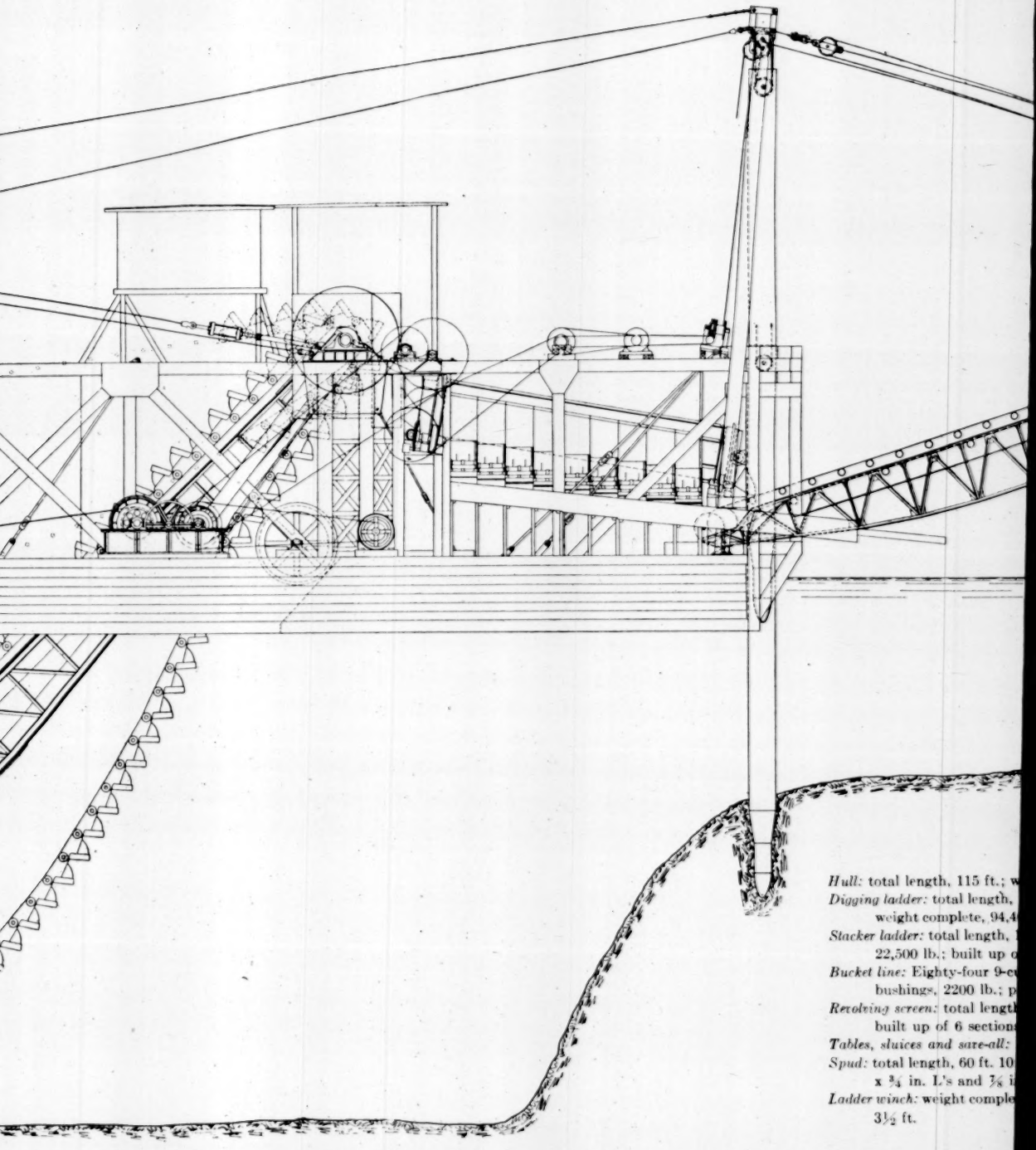


FIG. 4 PLAN, 9-CU. FT. DREDGE

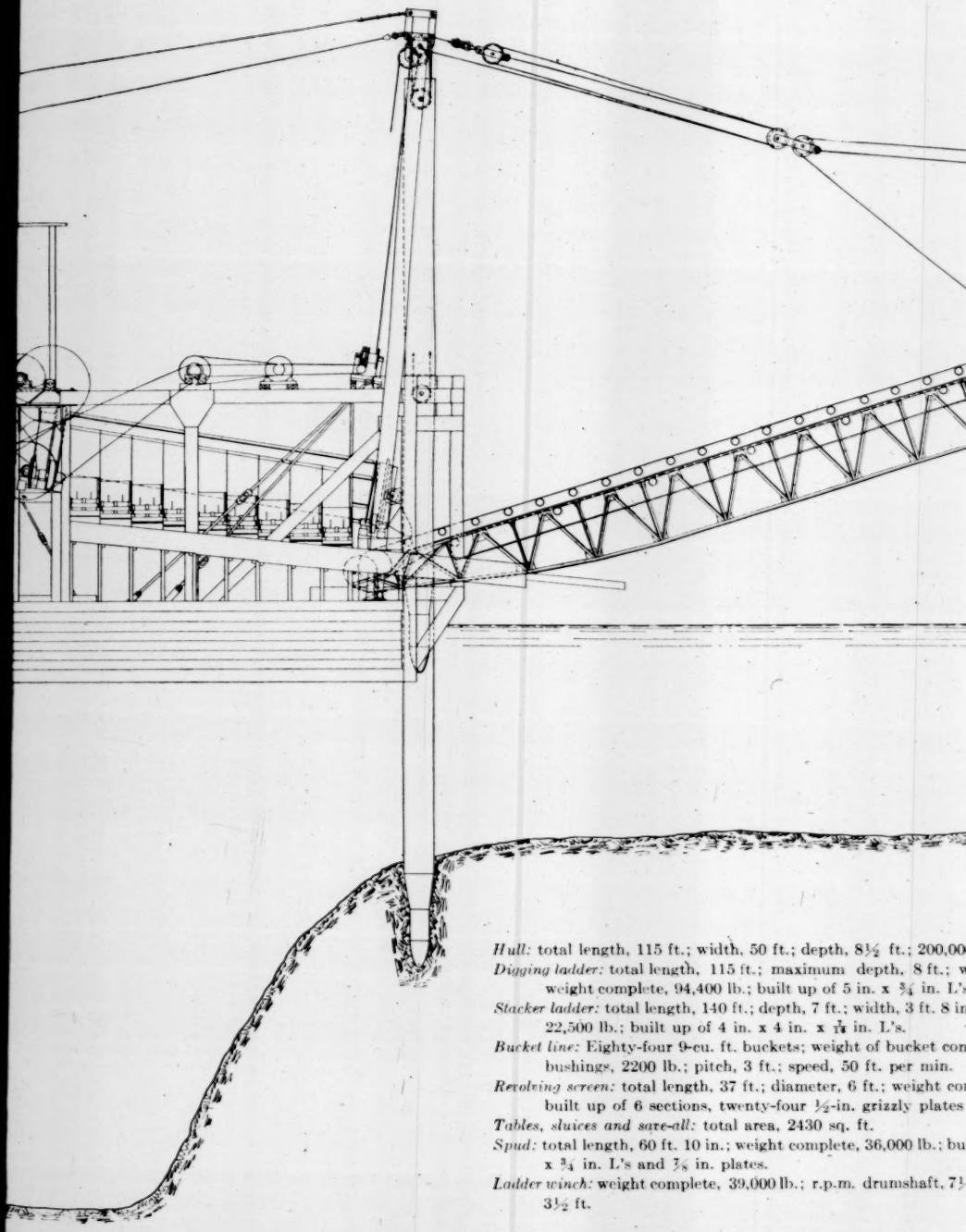






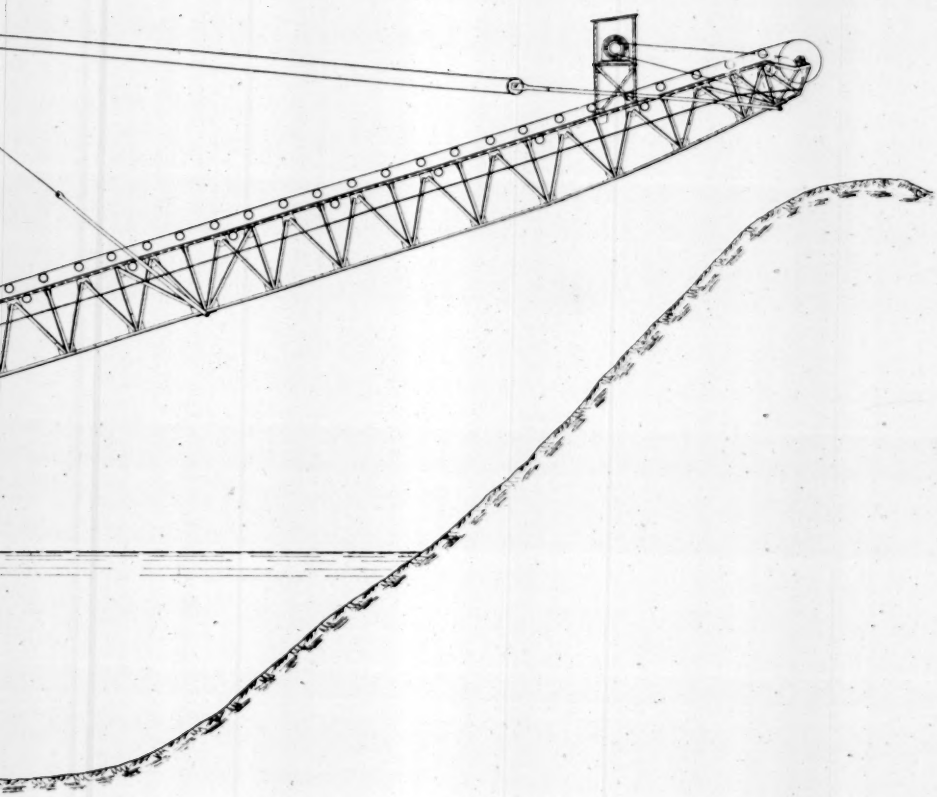
Hull: total length, 115 ft.; w
Digging ladder: total length,
weight complete, 94,4
Stacker ladder: total length,
22,500 lb.; built up o
Bucket line: Eighty-four 9-c
bushings, 2200 lb.; p
Revolving screen: total lengt
built up of 6 section
Tables, sluices and save-all:
Spud: total length, 60 ft. 10
x ¾ in. L's and ⅞ i
Ladder winch: weight comple
3½ ft.

FIG. 5 ELEVATION, 9-CU. FT. DREDGE



Hull: total length, 115 ft.; width, 50 ft.; depth, 8½ ft.; 200,00
Digging ladder: total length, 115 ft.; maximum depth, 8 ft.; w
weight complete, 94,400 lb.; built up of 5 in. x ¾ in. L's
Stacker ladder: total length, 140 ft.; depth, 7 ft.; width, 3 ft. 8 in
22,500 lb.; built up of 4 in. x 4 in. x ⅞ in. L's.
Bucket line: Eighty-four 9-cu. ft. buckets; weight of bucket con
bushings, 2200 lb.; pitch, 3 ft.; speed, 50 ft. per min.
Revolving screen: total length, 37 ft.; diameter, 6 ft.; weight co
built up of 6 sections, twenty-four ½-in. grizzly plates
Tables, sluices and sare-all: total area, 2430 sq. ft.
Spud: total length, 60 ft. 10 in.; weight complete, 36,000 lb.; bu
x ¾ in. L's and ⅞ in. plates.
Ladder winch: weight complete, 39,000 lb.; r.p.m. drumshaft, 7½
3½ ft.

FIG. 5 ELEVATION, 9-CU. FT. DREDGE



GENERAL DATA OF 9–CU. FT. DREDGE

200,000 bd. ft.
8 ft.; width, 4 ft. 7½ in.;
in. L's and ½-in. plates.
ft. 8 in.; weight complete,

ket complete with pin and
min.
ght complete, 37,360 lb.;
plates ⅜-in. holes.

lb.; built up of 4 in. x 4 in.

aft, 7½; outside diameter,

Main winch: weight complete, 23,275 lb.
Pumps: high-pressure, 2800 gal. per min., 50 ft. head; 10 in. discharge; 12 in. suction.
low-pressure, 4000 gal. per min.; 25 ft. head; 12 in. discharge; 12 in. suction.
primer pump, 2-stage, 2 in. discharge; 2½ in. suction.
Motors: digging motor, 200 h.p.; screen motor, 50 h.p.
main winch motor, 25 h.p.; stacker motor, 35 h.p.
high-pressure motor, 100 h.p.; low-pressure motor, 35 h.p.
primer pump motor, 10 h.p.
Lines: ladder line 1⅛ in.–6/19–plow steel hoisting rope 1035 ft. long.
spud line, ¾ in.–6/19–C. G. S. hoisting rope 553 ft. long.
stacker line ¾ in.–6/19–C. G. S. hoisting rope 500 ft. long.
swinging line 1 in.–6/19–C. G. S. plow steel hoisting rope 600 ft. long.

were used to hold the dredge up to its work, the end of the digging ladder moving in the arc of a circle with the spud as its center.

9 About this time several dipper dredges were built, but although some were more or less successful they did not prove to be as efficient as the endless chain type and were soon abandoned.

10 In 1899 a double-lift dredge was built at Oroville, and soon after a similar dredge was built on the American River, the latter, however, being of the single-lift type (Fig. 2), using close-connected buckets (every link in the chain being a bucket instead of links and buckets alternating) with spuds, no stacker, but having its main

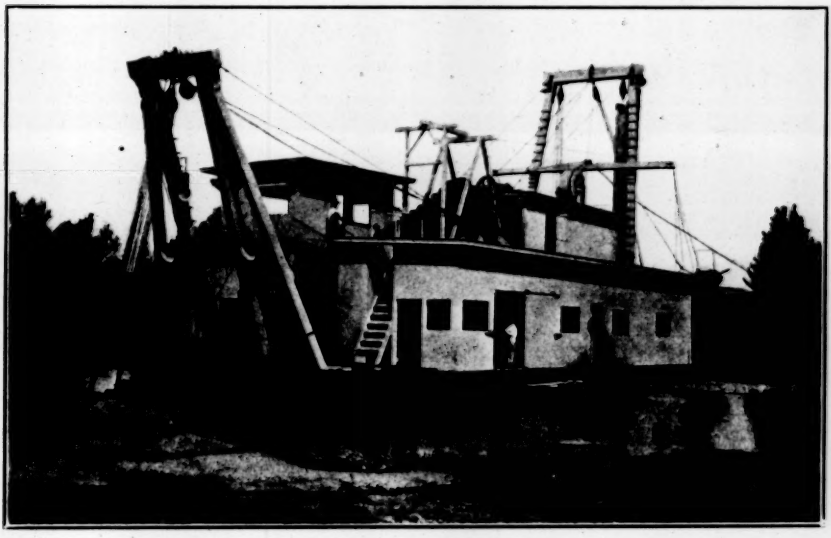


FIG. 3 FIRST CALIFORNIA TYPE GOLD DREDGE, 1901

gantry high, the buckets dumping into a revolving screen with large holes. This screen was raised high enough so that the finer material fell directly into a fore-and-aft sluice; the oversize dumped into stone shoots discharging on both sides of the dredge.

11 A judicious combination of the best features of the New Zealand and the double and single-lift types described above, together with several new ideas not used in any of the earlier dredges, resulted in a dredge being built some time in 1901, near Oroville, Cal. (Fig. 3), equipped with a shaking instead of a revolving screen, the holes in which were small, the fine material going to gold-saving tables as in the New Zealand type, but with Hungarian riffles replac-

ing the cocoanut matting and expanded metal. The oversize was stacked at the stern by means of a belt conveyor instead of a bucket

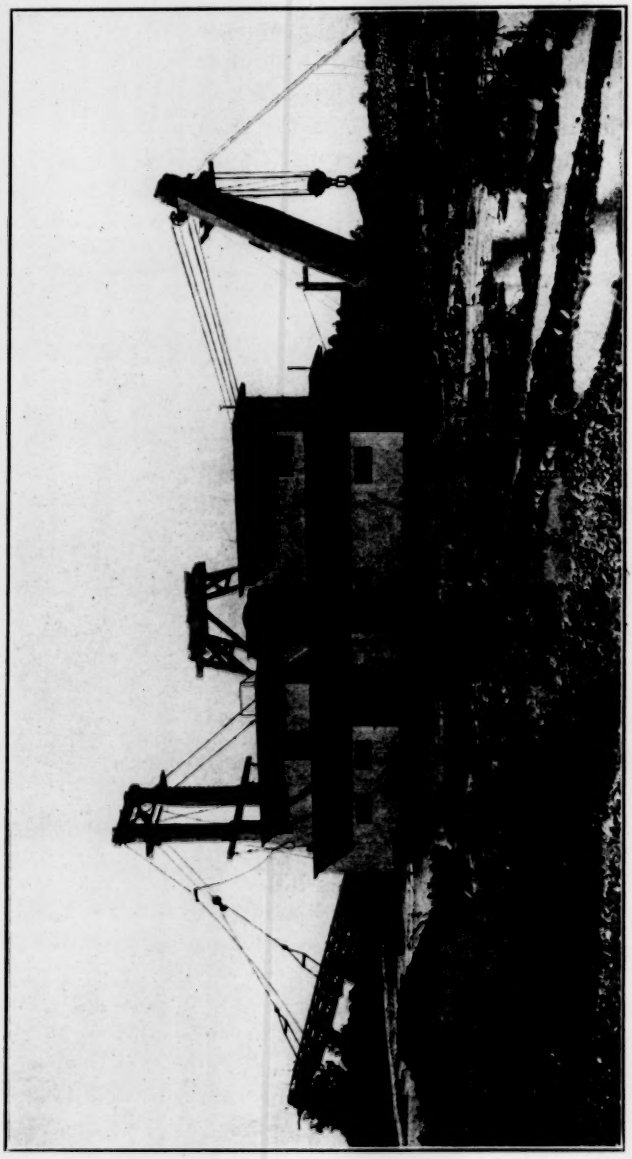


FIG. 6 MODERN 5-CU. FT. GOLD DREDGE, 1907

elevator. Spuds and close-connected buckets were used as in the single-lift type. This general arrangement became known as the

California type, and with its later modifications is the one to receive the most attention in this paper, and unless otherwise stated the descriptions are for this type of dredge. A plan and elevation of this type are shown in Figs. 4 and 5.

12 The original New Zealand type slightly modified is still sometimes used for the smaller, shallower and richer areas, where it is said to do very good work. It is cheaper to build and under certain conditions may prove a better investment than the more expensive California type.

13 The earlier dredges used buckets holding 3 cu. ft., then the 5-ft. size became numerous (Fig. 6), and later the 7, 8 and 9-ft. sizes (Fig. 7) were much used. Many went back to the revolving screen,

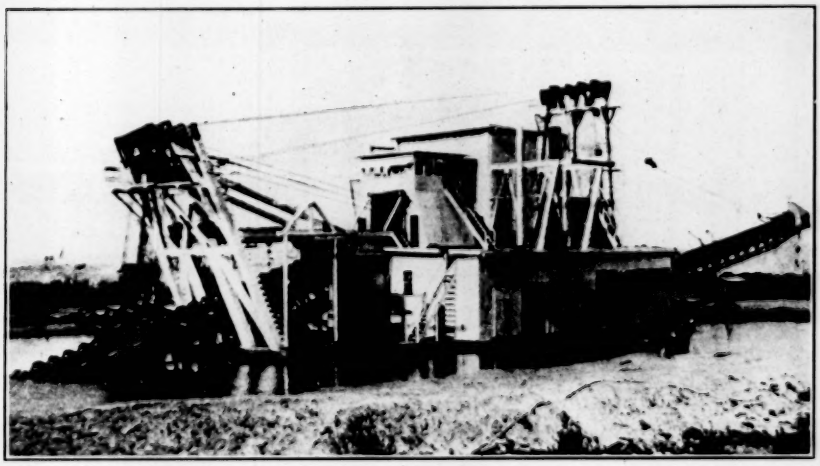


FIG 7 MODERN 9-CU. FT. GOLD DREDGE, 1911

noticeably in localities where much clay existed. In 1905 a 13½-ft. dredge was built near Folsom, Cal. (Fig. 8), and since that time several with 13½ to 16-ft. buckets have been put in operation (Fig. 9).

14 Having gained a general idea of the development of this type of gold dredge, its different component parts will be taken up in detail.

BUCKET

15 The bucket proper is made up of base, hood and lip (Fig. 10).

16 *Base.* The base is made of cast steel, many grades being used. It should be hard so as to stand wear, tough so as to stand the strain of digging, and should not be too expensive, since it has constantly to be replaced when worn or broken. All dredge costs

are figured on a basis of cents per cubic yard of gravel handled, so that the first cost is not vital so long as it will handle the requisite number of cubic yards before giving out.

17 The most common form of base has one hub or back eye fitting between the two front eyes of the bucket behind it. Some use two back and three front eyes. The advantage of the three-eye bucket is that a smaller diameter pin may be used, there being four planes of shear instead of two; the disadvantage is that the cost of machining is greater and there seems to be a tendency to crack these buckets between the two back eyes, owing probably to a prying

FIG. 8 MODERN 13½-CU. FT. GOLD DREDGE, 1905

action under hard digging conditions, particularly after the pins are partly worn.

18 Besides the ordinary high-carbon open-hearth steel, chrome nickel and manganese steels are used. One essential is that the bucket casting should be thoroughly annealed before it is machined. It is my opinion that many breaks in bucket bases have been caused solely by improper annealing.

19 A template having long guiding bushed holes is useful in drilling for the hood so that the rivet holes will be accurately spaced both inside and out. This is essential in order that there may be no lost time in making replacements. In machining the base the holes in the eyes should be very carefully bored so as to be exactly parallel with the hole in the hub. In order to do this the bucket may be fitted with its bushing, a shaft put through the hub, the bucket

brought tightly against it, and then the eyes bored parallel to this shaft.

20 *Hood.* The bucket hood is made either of pressed steel or cast steel. There are also buckets used which have the hood cast as an integral part of the base. The pressed steel hoods are made in either one or three pieces. The thickness depends on the size of bucket and the nature of the digging, and varies from $\frac{1}{4}$ in. to $\frac{1}{2}$ in. It is sometimes reinforced at the back where the greatest wear occurs. The cast-steel hoods are usually made about $\frac{3}{4}$ in. thick, have a stiffening rib running up the middle, and sometimes shoulders for the lip to rest on and a shoulder for the base to fit into.

21 The advantages of pressed over cast-steel hoods are their cheapness, lightness, and ease of repair if bent or broken. A disad-

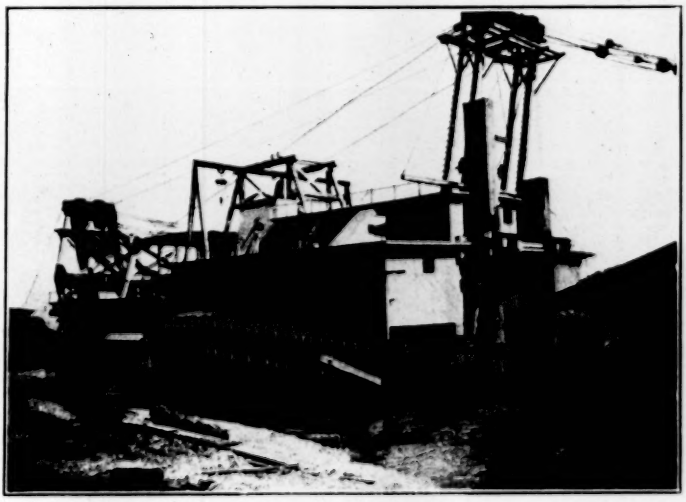


FIG. 9 MODERN 16-CU. FT. GOLD DREDGE, 1911

vantage is that the strength cannot always be put where wanted except at extra cost.

22 A standard drilled base may be used as a template for drilling the hood. The author considers the one-piece preferable to the three-piece hood because (*a*) it will stay water tight; and (*b*) there are fewer rivets to work loose and replace. The disadvantage is the difficulty of pressing a single piece into the proper shape, making the cost slightly greater than if three pieces were used.

23 *Lip.* Bucket lips are now nearly all made of cast manganese

steel. At one time the author used a very hard high-carbon nickel-steel forging, its only advantage being that it could be shaped on the ground to fit several different hoods and bases which it was desired to use up. A record was kept of about 100 of these lips and it was found that for cost and life they were very close to the manganese steel. For use in out of the way places and in emergencies they do

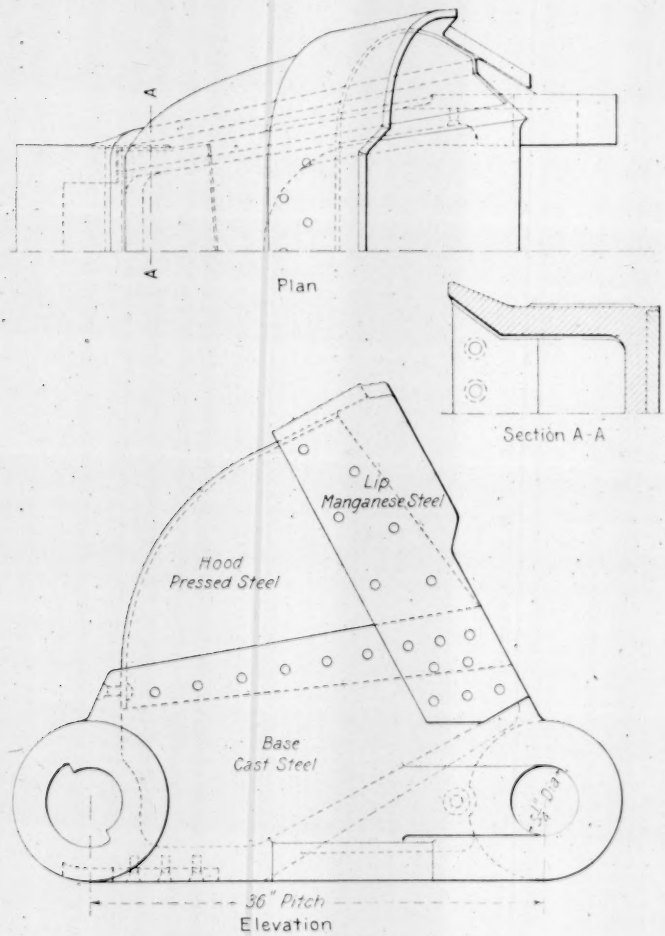


FIG. 10 9-CU. FT. BUCKET

very well. Their greatest disadvantage is that they cannot be readily forged into the more intricate shapes now commonly used.

24 The lips in use vary considerably in design according to the nature of the ground and the fancy or judgment of the designer.

In the first place, a lip must be made sturdy enough to perform its work without bending or breaking, even up to the time it is ready to be replaced. If it is bent or broken before this time, damage is done the hood, with the resultant extra cost of repairing the hood as well as replacing the lip. The thickness of the lip is determined by the above conditions, but its width is a matter of judgment, since if made too wide the capacity of the bucket is materially reduced by the time it is worn out; if too narrow, it must be replaced often and a large loss made in each replacement both in time and unused metal.

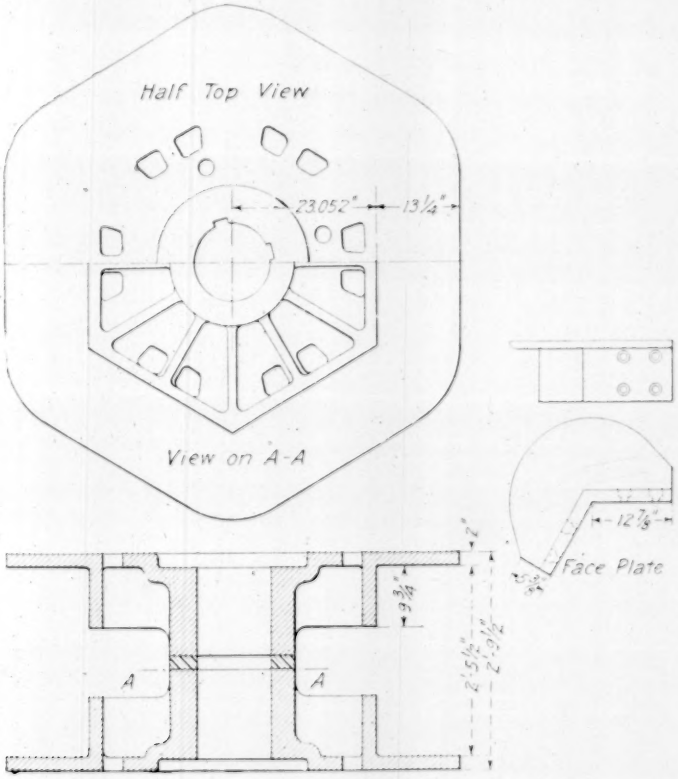


FIG. 11 LOWER TUMBLER FOR 9-CU. FT. DREDGE, SHOWING WEARING PLATE

25 Some engineers prefer a highly arched digging face, others a rather flat face with sharply rounded corners. Those favoring the arched face contend with good reason that it is a stronger shape, and those favoring the flat face and rounded corners think that this shape is a better digger. The shape of the digging edge is another matter in which designers differ; some make it perfectly straight, others

prefer corner extensions at the points where the wear indicates most of the digging is done. The author is in favor of the corner extensions, since he believes they dig better and the lip is more completely worn out when it is ready to be replaced.

26 The fewer rivets used the smaller will be the cost of re-assembling, but care must be taken to have enough rivets to do the work without shearing or pulling out. On many of the lip designs a shoulder is cast so that the hood edge fits against it, thus taking much of the strain off the rivets.

27 *Pin.* There have been many attempts to improve the design of bucket pins, those ordinarily used being rather expensive to make. It is a round shaft with a flat lug or "L" at one end. The L fits into a slot machined in the bucket base and prevents the pin turning in the eyes. Dowel pins, keys, squared and slotted ends, tee shapes and many other devices have been tried, but so far as the author is aware none of them has proved as satisfactory as the old L design.

28 The pins are made of various kinds of steel, the object being to use a very hard material which is also tough and not too expensive. A high-carbon steel, low in sulphur and phosphorus, forged, machined, and then oil tempered, is most common. Various forms of nickel, chrome and manganese steel are also used, but so far have not become general practice.

29 *Bushing.* The bushings are made in pairs, half round, and fit a recess in the bucket hub, placed at such an angle that the wear will be evenly distributed. They should be a snug fit, but $\frac{1}{4}$ in. to $\frac{1}{2}$ in. short on the ends to allow for the flow of the metal. In using manganese steel this tendency to flow must always be taken into consideration and suitable provision made for it in the dimensions.

30 *Insert Plates.* During the last few years manganese steel insert plates have come into general use in order to protect the bottom of the bucket base near the hub, at which point excessive wear occurs on account of the slip on the upper tumbler. Often buckets are found to be worn out at this point but perfectly good everywhere else. The manganese plates are of such a shape that after they are put in place and the softer steel of the base peened over the edges they retain their position without other means of fastening.

LOWER TUMBLER

31 At the lower end of the digging ladder the buckets pass over a 6-sided flanged tumbler (Fig. 11) which revolves between forks at-

tached to the digging ladder. It is cast of open-hearth steel. Wearing plates are used to protect the faces and the flanges. The detail of design varies on different dredges. Some use a tumbler having curved extensions at the corners of the flanges, others have the corners rounded but not extended, and still others use a circular flange. The extended corners require less metal but are likely to be broken off, and the circular flange may interfere with the digging. The design in which a side view will show a hexagonal shape with rounded corners is strong and cannot well interfere with the digging. The height of flange is another point of difference and varies from 6 in. to 13 in. at the corners. It must be made high enough to keep the buckets from running off, but if too high may interfere with the digging. It is also expensive.

32 The slope or flare of the flange varies from 0 to 1 in 10. It is claimed by some that if the sides are made too straight that the buckets are likely to run off and if the flare is too great the eyes of the buckets will wear thin at the bottom. I do not know that any standard of flare has been adopted, but I am inclined to the belief that a straight or nearly straight flange is best, all things considered.

33 *Wearing Plates.* The flanges and tumbler faces are protected by wearing plates (Fig. 11) made of some hard steel, nickel, chrome and manganese all being used. So far as wearing qualities are concerned the manganese steel gives the best result, but this material is so hard that it cannot be machined and it is difficult to make the plates fit the tumbler accurately; also, since the rivet holes have to be cored they are likely to be a bad fit. Besides these disadvantages the manganese steel plates are very destructive to the bucket line and it is better to wear out tumbler plates than buckets. Several tumblers have recently been made with a manganese steel disc set into a recess at each corner and held in place by peening over the edges in a manner similar to the bucket insert plates described above. This method is said to have given very good results. Some use separate plates for protecting faces and flanges, while others have the wearing plates made so as to protect both faces and flanges. The advantage of the last method is that the plates are very strong and not likely to break or work loose.

34 *Shaft.* The lower tumbler shaft is usually made of nickel steel, either oil tempered, annealed, or both, and may be hollow bored or solid. It should be very securely keyed to the tumbler, and the author has found that four taper keys, two at each side, 90 or 120 deg. apart, have given the best results. Provision should be made for

driving the keys out from the inside and the tumbler should be either shrunk or pressed on.

LADDER ROLLER

35 The ladder rollers (Fig. 12) serve to support the buckets on their way from the lower to the upper tumbler. Cast iron was formerly used, but now cast steel, either high carbon, chrome or manganese, is most in favor. Very hard chilled white iron cast around a soft core has given very good results; also manganese steel made in the same way has proved even more efficient so far as wear is concerned. The writer's objection to the manganese roller is that after considerable use the soft core, if made thin, is likely to become

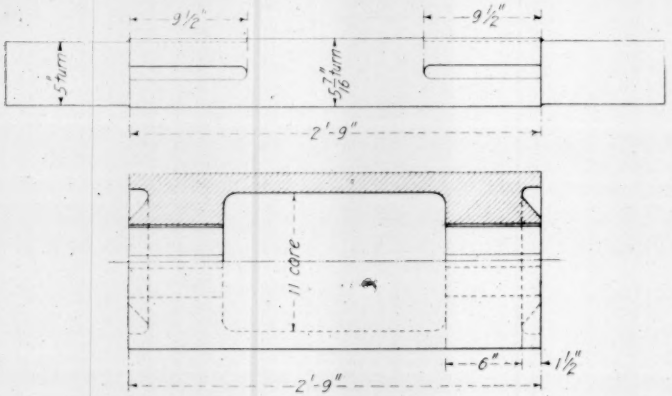


FIG. 12 LADDER ROLLER FOR 9-CU. FT. DREDGE

loose, and if made thick the difference in expansion of the two metals will cause the outside shell to crack.

36 A high-carbon or chrome-steel casting makes a good roller, and can be made just soft enough to machine; it will then wear well and need not have a soft iron core. Particularly good results have been secured by the author with chrome-steel rollers made in this way.

37 *Shaft.* There has always been trouble in keeping shafts tight in the rollers. The writer uses two taper keys at each end, 90 deg. apart, the same plan as for the lower tumbler shaft. This method of fastening has proved perfectly satisfactory.

UPPER TUMBLER

38 There have been many forms of upper tumbler tried, but now

the common practice is to use a 6-sided plain casting, onto which is bolted a set of heavy cushion plates with guides cast on the sides (Fig. 13).

39 *Cushion Plates.* These plates are made to lap over the corners and fit into transverse slots in the tumbler casting. This is done to prevent them from becoming loose, as was the case before this design was perfected. They are made of cast steel, either high carbon, chrome, nickel, or manganese. The cushion plates are bolted to the main tumbler casting by means of large bolts $1\frac{1}{2}$ or 2 in. in diameter with square countersunk heads on the outside (Fig. 14).

40 *Wearing Plates.* Wearing plates are used on the cushion

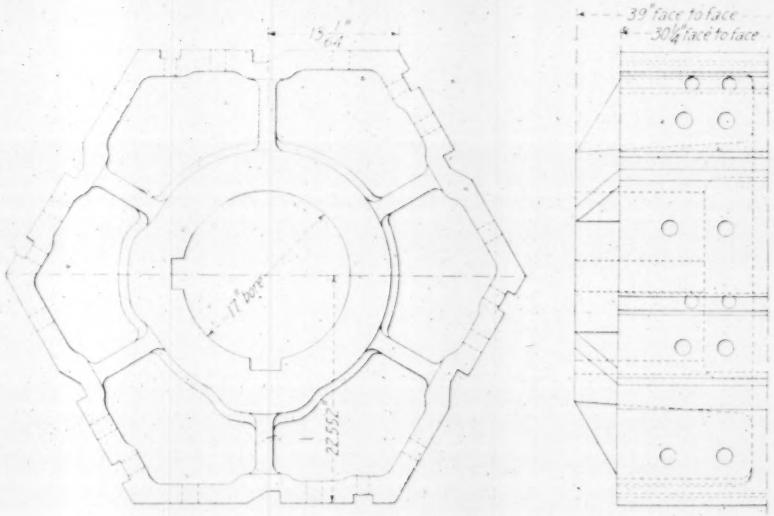


FIG. 13 UPPER TUMBLER FOR 9-CU. FT. DREDGE

plates, both for the faces and the guide flanges on the sides. These plates may be made of any material desired, but the same objection to manganese exists on the upper as the lower tumbler. The author uses plates forged from worn-out bucket pins and then gives them an oil temper. These have given good results, but care should be taken not to temper them too hard, since they then become brittle and will give poor service. The wearing plates are riveted with countersunk rivets to the cushion plates. Riveting is better than bolting for this purpose.

41 *Shaft.* It is imperative that the upper tumbler shaft should be of ample strength and be made of excellent material since it has

to bear the brunt of the digging. A factor of safety of not less than 10 is recommended, and it would do no harm to have it even higher. It is customary to use a material of high tensile strength, since if made of low tensile strength material the shaft will have to be so large that it will interfere with the design of the upper tumbler which it carries. Nickel steel, hollow bored, oil tempered and annealed shafts are much used, having a tensile strength of 90,000 lb., elastic limit 60,000, elongation 22 per cent, contraction 50 per cent. Excellent results have also been obtained with Krupp crucible steel shafts, hollow bored, oil tempered and annealed, having a tensile

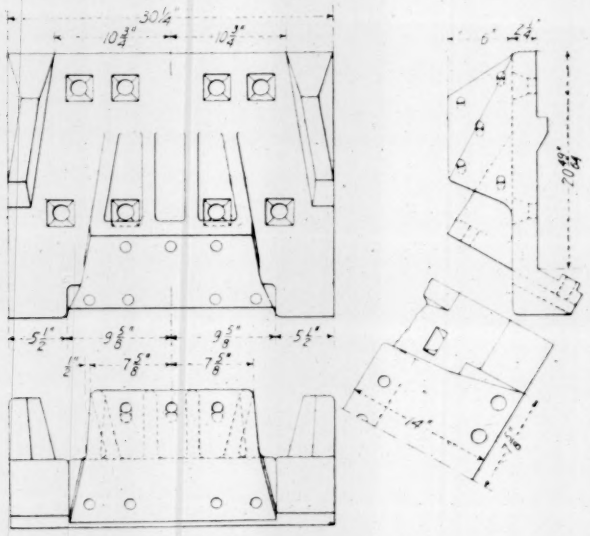


FIG. 14 CUSHION PLATES FOR UPPER TUMBLER, 9-CU. FT. DREDGE

strength of 80,000 lb., elastic limit, 43,000, elongation 22 per cent, contraction 45 per cent. Lately the author has used a nickel steel solid shaft which has been thoroughly annealed but not oil tempered. This has given excellent results and is inexpensive. The physical properties are as follows: Tensile strength, 80,000, elastic limit 50,000, elongation 25 per cent, contraction 45 per cent. A shaft of soft iron or steel, having a low tensile strength, but great toughness, such as Lloyds' test steel, would be the ideal shaft for this purpose were it not for the necessity of making them so large that there is no

room left to fasten the cushion plates to the upper tumbler. By using a 7 or 8-sided tumbler this difficulty could be overcome.

DIGGING LADDER

42 The design of the digging ladder varies with the depth and nature of the ground to be handled. It is either a plate girder or a lattice girder or truss (Fig. 15).

43 The plate girder type is the most common and in the writer's opinion the best. A little weight may be saved by using the lattice type, and although it may be as strong as a plate girder, there is in the lattice type more danger of rivets working loose, causing ultimate collapse.

44 Ample space should be allowed under the ladder rollers, otherwise the spill from the buckets will collect and prevent the rollers turning. The ladders are made in several sections so that they may be more easily transported. Where the splices occur great care should be taken to make the joints of ample strength. The author has seen several digging ladders fail and in nearly every case the failure occurred at one of the splices. In figuring the strains in a digging ladder it should be considered not only as a beam supported at two or three points, but also as a column. Where a single line is used for side swinging, the strains must also be figured for side as well as vertical stress.

45 *Lower Forks.* At the lower end of the ladder cast-steel forks are riveted to the structural portion of the ladder. These forks are made differently by various designers, but their duty in all cases is to carry the lower tumbler bearings and give a space for the lower tumbler to revolve in. These forks should have great lateral strength since there is a tendency to spread under the severe digging conditions. The writer has found it of advantage to make the inside edges of these forks fit very closely to the outside of the tumbler, since if a large space is left rocks lodge and grind out both the tumbler and the forks. In his last design he has put in replaceable manganese steel liners on the inside of the forks and hopes that it will prevent much of the wear on tumbler flanges.

46 *Lower Tumbler Bearings.* There are several designs of lower tumbler bearings used. In the most common the lower portion of the forks are made to form one-half of the bearings and to this the other half is bolted. In this design a replaceable cast-iron bushing is made to fit inside the main casting, often of the self-aligning type. A design similar to that of the locomotive connecting-rod end, which

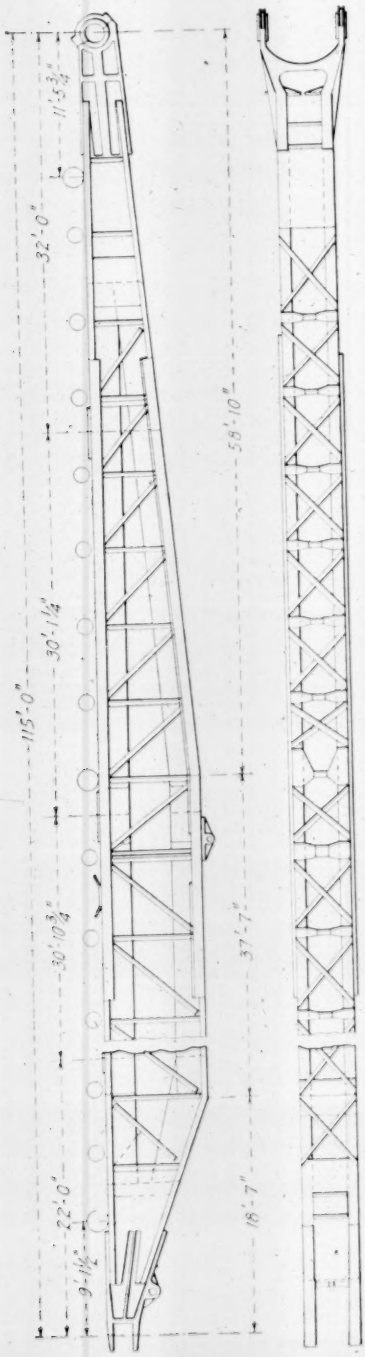


FIG. 15 DIGGING LADDER FOR 9-CU. FT. DREDGE

is fastened and adjusted by means of taper keys, has been used by the author. The bearing is made of cast iron in two halves and slides between the upper and lower extensions of the forks. Square holes are machined through these extensions to take the keys. The half bearings are interchangeable and can be used for the upper or lower half and for either side. This style of bearing has been found very satisfactory, since it is cheap and easily replaceable.

47 Many attempts have been made to use protective devices for keeping grit out of the bearings, rubber, felt, brass and iron rings having been tried. The rubber and felt have been entirely discarded. Some still use bronze or iron protective rings, but they are all more

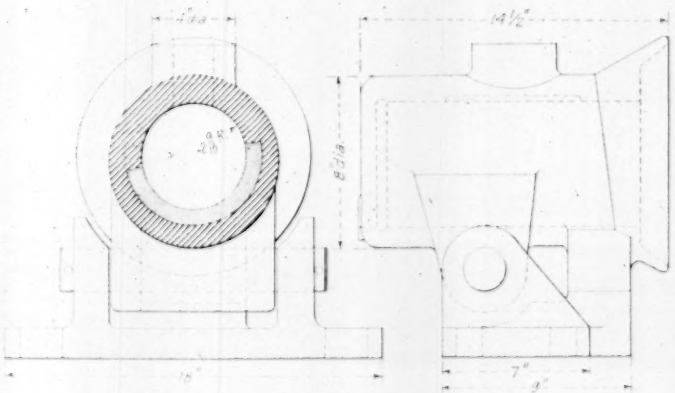


FIG. 16 LADDER ROLLER BEARING FOR 9-CU. FT. DREDGE

bother and cause more delays than they are worth. A hole through which grease may be introduced is provided at an easily accessible place on the bearings. The grease is introduced by means of a grease gun with a screw-operated piston.

48 The writer uses a lower tumbler shaft with shallow flanges turned at either end. These flanges fit into recesses on the outside of the tumbler bearings, thus preventing the ladder forks from spreading. Where hollow shafts were used it was common practice at one time to put a rod lengthwise through the shaft, a nut at either end preventing the forks from springing apart. With the modern practice of making the lower forks extremely massive there is little danger of springing and precautions need not be taken in this direction.

49 *Roller Bearings.* Spaced along the digging ladder at regular intervals are the roller bearings supported in chairs by pins which

allow movement in a plane at right angles to the length of the ladder (Fig. 16). The first bearings used were fixed rigidly and much trouble was experienced by breaking roller shafts, probably caused by movement in the ladder truss. I think it likely, with the heavier ladder construction now in vogue and the larger-sized roller shafts, that the fixed bearing would give little trouble; but even so, the present common design is better since the bearing may be very quickly and easily removed, and this is a desirable feature in any part of a dredge.

50 The roller bearings are made with half-round bushings similar to those used in the bucket hubs. Cast iron, manganese steel and bronze are used for this purpose. Cast-iron roller bearings are being replaced by cast-steel. Where a self-adjusting bearing is used it should be supported near the middle and be of ample length, otherwise it will tend to tip down and wear out unevenly and rapidly.

51 *Catch-all Pan.* In some of the dredges a pan is provided below the rollers which conveys the slop-over of the buckets to the lower end of the ladder where it will be picked up again. This is a very good idea, but care should be taken to make it at least 12 in. below the rollers and in small, easily replaceable sections. If not deep enough it is likely to be a source of much annoyance by filling up and stopping the rollers, or if made in long unprotected sections it will wear out and require much work and loss of time in replacing.

52 *Suspension.* The lower end of the ladder is supported at either one or two points, the single suspension being used for the shorter ladders and the two-point suspension for the longer ones. Several different designs are in use, the most common being a double set of blocks, one on each side, attached to heavy forged rods or bars connecting with the ladder. Heavy chains, forged or cast links and pin links are also used. Another type has a single set of blocks attached to a cross-piece, the connection to the ladder being made from each of its ends. The attachment to the ladder is usually made by means of a cross-shaft or pin running through bearings fastened to the under side of the ladder.

53 *Upper Forks.* The upper end of the digging ladder is pivoted either on the tumbler shaft-bearing support or on a separate mounting entirely detached from the bucket drive. By employing the first method the tumbler shaft may be used as a center and the relative position of the bucket line and upper tumbler will remain constant for all depths of digging. In the second method the pull on the bucket line will be at different angles according to the position of the ladder. The first method, therefore, has a decided advantage, but

it also possesses the disadvantage of preventing the easy removal of the upper tumbler and shaft, is more expensive, and in case of accident to the ladder the bucket drive may be involved through the tumbler shaft-bearing support. Massive cast-steel forks are used when the ladder is pivoted on the tumbler-bearing supports, and they should extend well down the structural part of the ladder in order to give ample space for riveting.

54 *Chafing Beams.* Along the sides of the ladder wooden chafing beams are bolted, the outer faces of which are protected by steel plates. The author prefers beams put at both top and bottom of the ladder so that they will steady it when digging hard. The beams on the ladder rub against similar ones on the sides of the well. Those in the well are put either at right angles to the ladder at intervals, or one may be placed on each side of the well parallel to the deck and on the streak above water line. If this form of chafing beam is used both the ladder and the well beams should have beveled timbers fitted above them so that rocks will not lodge and cause trouble when the ladder is raised or lowered. The disadvantage of placing the chafing beams at right angles to the ladder is that in case anything catches on one of them near the bottom and tears off a plate it is very difficult to repair, while if the beams are entirely above the water line they are easily accessible. The only disadvantage of placing them in this way is that the distance from the contact point of ladder and beam to the ladder's end is greater by the draft of the hull, and in case of side strain the bending moment is greater. Where a shore sheave is used and the end of the swinging line is attached to the lower end of the ladder there is very little side strain, and in this case the horizontal wearing plates are best, in the writer's opinion. Where a single swinging line is used, running directly to the shore, considerable side strain is exerted on the ladder, and the chafing beams at right angles to the ladder are preferable. Some designers provide rollers for the sides of the ladder, which serve the same purpose as the chafing beams.

BUCKET DRIVE

55 The buckets are driven by a chain of gears mounted on the main gantry. The use of a chain or shaft drive from the lower deck has now been entirely abandoned in California practice. Some dredges drive direct from the digging motor to the pulley shaft on the main gantry. Others use a countershaft on the main deck, from which also is driven the ladder hoist. Neither type has any great

advantage over the other, and if properly designed either method will give satisfactory results.

56 The upper tumbler may be driven by a gear at one end of the tumbler shaft, or one at each end, with a double set of intermediate pinions, gears, and pulley-shaft pinions. The single drive is of course cheaper and simpler, but experience has shown that after many years of service the single drive is inclined to get out of line, even

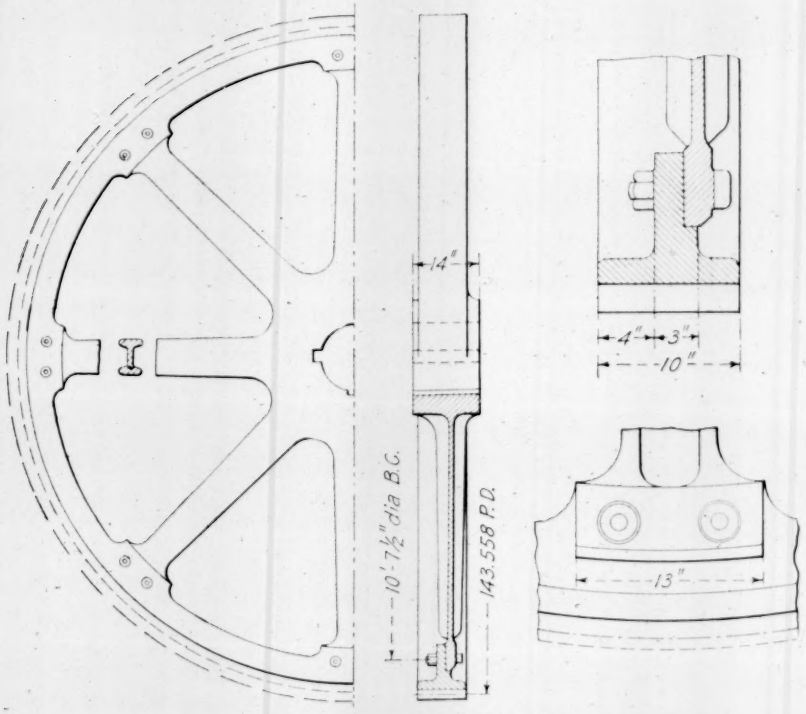


FIG. 17 BULL WHEEL FOR 9-CU. FT. DREDGE

when very solidly built. The writer very much favors the double drive and would advise its use.

57 *Bull Wheel.* The bull wheels or tumbler-shaft gears are cast of steel in sections, sometimes with a split and sometimes with a solid hub (Fig. 17). The rim is cast in one piece or in halves. Some designers use cut teeth, others rough cast. There can be no doubt but that the cut teeth run more smoothly, but for the rough work in hand the writer does not believe that the extra cost is warranted,

particularly since the teeth are soon worn down and the benefit of the cut teeth disappears. In order to prevent breaking, all sharp angles should be avoided in the design of the spokes. They are subjected to constant shocks and vibration, and the gears will invariably break at a sharp re-entrant angle should such occur at a vulnerable point. In order to make a snug fit and also to keep the rim from becoming elliptical, the spokes should either have a shoulder fitting on a machined inner edge of the rim, or the ends of the spokes should be turned to fit a machined shoulder on the rim. The latter method is preferable since it does away with sharp corners which, as explained above, are always points of weakness.

58 It has been common practice to use one large bolt to fasten each spoke to the rim. Two smaller bolts are better since a more even distribution of spoke cross-section can be maintained by this design and it gives a longer spoke end to bear against the machined rim. A good many spokes have been broken near the rim where the cross-section is ample but where a point of weakness exists owing to the re-entrant shoulder above mentioned.

59 *Intermediate Shaft Pinions.* The intermediate shaft pinions are made solid and as small in diameter as is consistent with good design. These pinions wear out quite rapidly and should be so arranged as to be easily replaced.

60 *Intermediate Shaft Gears.* The intermediate shaft gears are designed of a size to give the bucket line the desired speed. They are often made with separate rims so that they may be replaced when worn without throwing away the rest of the gear.

61 *Pulley-Shaft Pinion.* The pulley-shaft pinions are made similar to the intermediate shaft pinions and as small as good design will allow.

62 *Clutch.* Whether the digging motor is belted direct to the bucket-drive pulley shaft, or to a countershaft on the ladder-hoist foundations and from there to the bucket-drive pulley shaft, some form of friction clutch is used to connect and disconnect the bucket drive, and for purposes of safety in case the buckets encounter obstructions beyond their strength. Any of the standard forms of friction clutches may be used, but the author prefers the electromagnet clutch and uses it in all his designs. It is by far the simplest to operate, is positive and will slip at exactly the same pull every time according to the amount of current sent through its coil by the rheostat controlled from the pilot house.

63 The clutch used is very simple in design (Fig. 18). A cast-

iron pulley is made so that there is a flat ring or web joining the spokes at about half the distance from hub to face. To this web is bolted a soft cast-iron ring. Between this and the pulley are inserted two thin sheets of copper and one of soft iron between the copper plates. These ring-shaped sheets are used in order to keep the lines of force where they are most wanted and to prevent the current making a permanent magnet of the pulley, shaft, etc., which would cause the clutch to drag when released. The pulley has a long hub

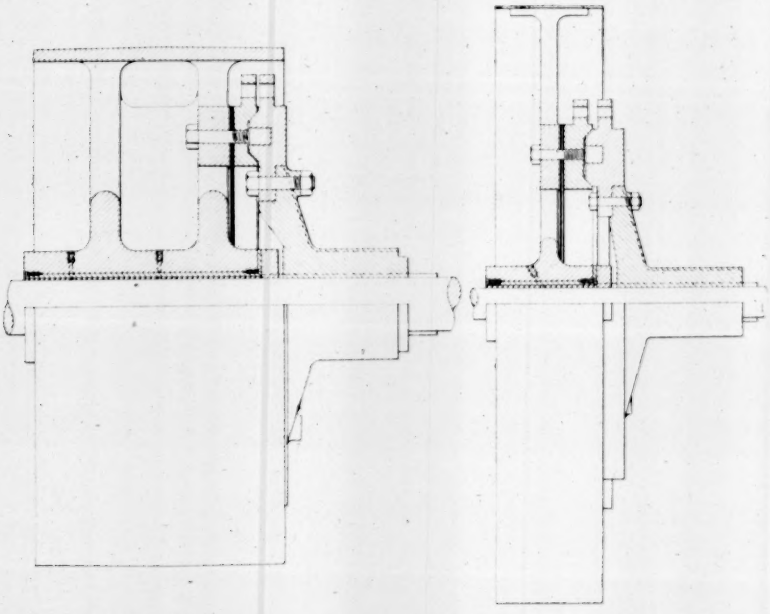


FIG. 18 MAGNETIC CLUTCHES FOR BUCKET DRIVE AND MAIN WINCH 9-CU. FT DREDGE

bushed with bronze and runs free on the shaft. A long cast-iron hub with two feather keyways at 180 deg. and a web extending out to the same diameter as the ring on the pulley is made to slide along the shaft. Onto this web is bolted another soft iron ring similar to the one bolted to the pulley. An annular groove of rectangular cross-section is machined in this ring midway between its inner and outer circumference into which fits a coil of magnet wire. By introducing an electric current into this coil through controller rings, magnetic lines of force are induced which will draw the two soft iron rings together, the pressure depending on the amount of current put through and the number of turns in the coil. The edges of the soft iron rings

are beveled so as to make a female joint on the pulley and a male joint on the sliding hub. The steeper the slope of these edges the less current will be required to do the requisite work, but if the cones are made too steep it will be difficult to disconnect the clutch. Forty-five degrees does very well. Good results are given by 440 turns of No. 16 magnet wire, and 4 amperes at 110 volts are put through the coil when working average ground. These figures are for a 200-h.p. clutch.

64 *Shafts.* The shafting for the bucket drive should be made of ample dimensions and of good material. Nickel steel annealed and a factor of safety of 10 are used by the author. The gears should be fitted very neatly and securely keyed.

65 *Bearings.* The upper tumbler-shaft bearings are made on an angle so as to have the bearing cap at right angles to the average line of pull of the bucket line. Cast-iron liners are sometimes used, but the author prefers a babbited bearing. The bearings should be well fastened so that they cannot slide forward, there being a very heavy pull in this direction. These bearings, like the others of the bucket drive, are made of cast steel.

66 The intermediate and pulley-shaft bearings are made of the ordinary type of babbited base and cap, but are heavier than standard and use very large diameter bolts, which should be made of a soft grade of iron or steel, since unless this is done they are likely to crystallize and break.

67 *Brake.* It is customary to have the pulley shaft extended and a brake wheel keyed on. This brake is operated from the pilot house and is used by the winchman to hold the buckets from going over the upper tumbler when the bucket line breaks, to stop quickly in case of an accident, and to hold the bucket line when making repairs or changing buckets.

68 *Pulley.* Standard medart pulleys are generally used. They give considerable trouble on account of broken spokes. The author has recently used with success a wooden bucket-drive pulley similar to that commonly employed in stamp mills, and on the last boat he designed a medart pulley with spokes made extra heavy near the hub and tapering to a little more than standard cross-section at the rim, since nearly all the trouble arose from spokes breaking near the hub. If a cast-steel hub and spokes could be made it would be the best pulley for the purpose. Several of the steel foundries say that they would not care to attempt to cast a spider with as small spokes as the standard medart and that a very heavy spoke would have to be

made in order to overcome shrinkage trouble. If spokes could be made of cast steel and of as light or lighter design than the standard medart it seems as if it would be an ideal pulley for dredge use.

69 *Foundations.* Too much attention cannot be given to making the supports for the bucket drive of very sturdy construction. In the first place these foundations are 20 or 30 ft. above the main deck and are subjected to constant and racking strains. If any portion of

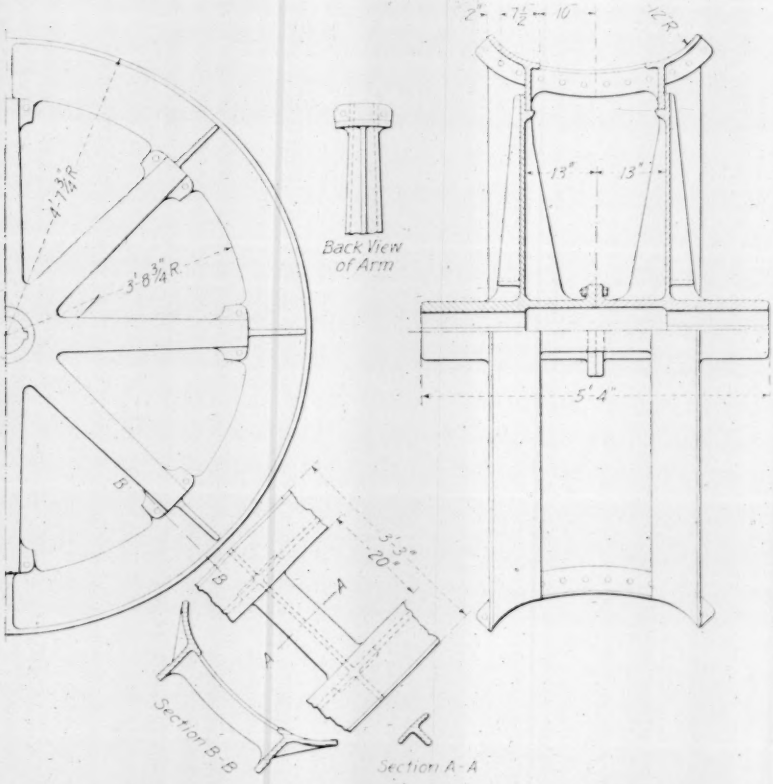


FIG. 19 BUCKET IDLER FOR 9-CU. FT. DREDGE

these supports gives it throws the shafting out of line with obvious results. One form of support now much used and considered very good has the upper cord of the main truss made high enough to take the upper tumbler-bearing supports directly on its top. By this method the main gantry is braced fore and aft in a most excellent manner and it is a much more rigid construction than where the

gantry extends above the cord. Large steel castings are used to tie the timbers together and to support the bearings directly. The details vary on different dredges and apparently have all given good service. One point which should be carefully looked into when designing this part of the dredge is so to place all bolts that they may be readily accessible and replaceable.

BUCKET IDLER

70 Where the depth of gravel to be dug demands a long string of buckets, an idler (Fig. 19) is used to support the sag. This idler is located at the throat of the well as low down as convenient. It is made up of a cast-steel hub and a double set of spokes bolted onto a wide rim made to fit the curve of the bucket lip. The shaft is keyed to the hub and revolves in two bronze bushed bearings set on the deck or other convenient place.

CATCH-ALL

71 Below the dump hopper an inclined set of grizzly bars is put in. These extend as far forward as the sag of the buckets will allow. The bars are put $\frac{3}{4}$ to 2 in. apart. Below this grizzly is a collecting pan which directs the gravel to the forward end of a sluice with riffles, occupying the space between the gantry posts and sloping toward the after portion of the dredge. This sluice is run back as far as the grade will permit and then dumps its contents into riffled side sluices running forward one on each side of the well. These discharge into the well 15 or 20 ft. forward of the well throat. Water is furnished from one of the pumps and washes the material dropping through the grizzly bars.

DUMP HOPPER

72 The hopper into which the buckets dump diverts the gravel into the screen. It is made either with a steel lining or by having a depression left at its bottom which fills up with rocks and gravel, thus forming its own lining. It is obvious that the last method is much more economical than the steel-lined hopper and that so far as its bottom is concerned it will last indefinitely; the only difficulty is that unless the hopper is left of ample dimensions it is likely to clog up when digging large quantities of material, particularly clay. By using a steel lining for the bottom of the hopper a little space may be saved, and if a large quantity of clay is to be handled the latter method might be preferable, but for ordinary conditions the rock bottom seems to be all that is necessary.

73 The hopper is built up on three sides of the upper tumbler, the top being at about the level of the bucket lips as they go over the tumbler. The forward upward edge is built to such a height as to

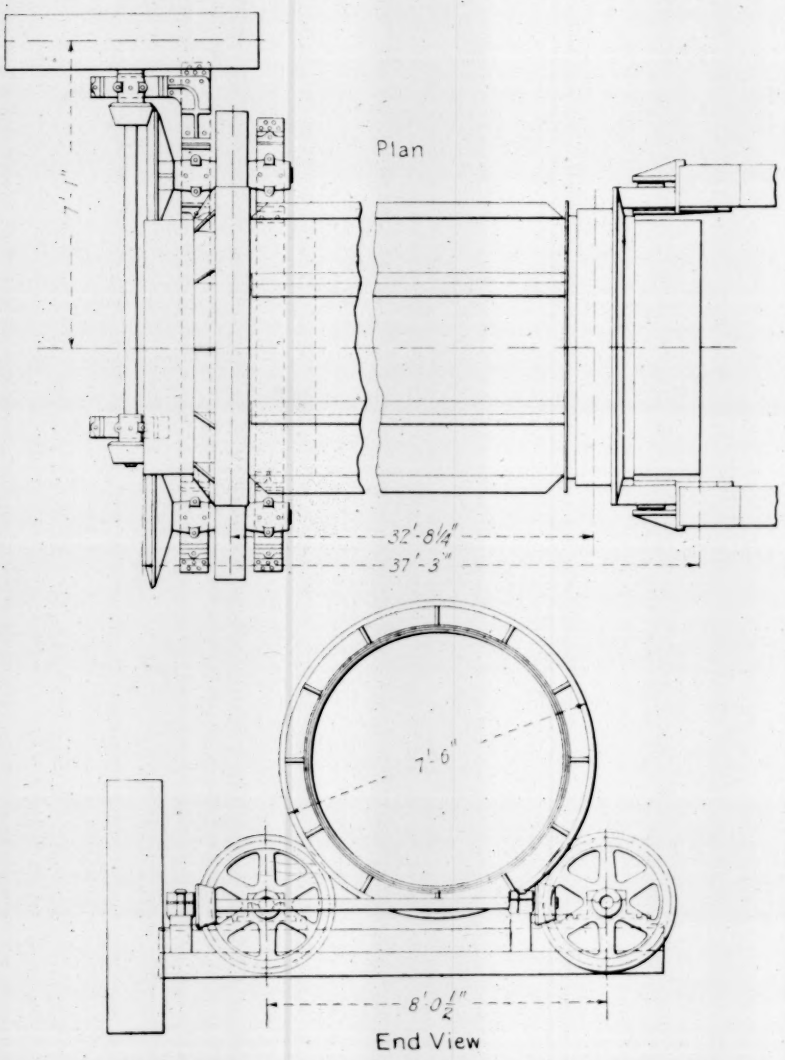


FIG. 20 REVOLVING SCREEN FOR 9-CU. FT. DREDGE

miss the bucket lips by a few inches when the ladder is in its lowest position, and it is of advantage to put inclined guide plates on each side of the bucket line to divert the slopover of the buckets into the

hopper below. The rear vertical wall of the hopper is subjected to considerable wear by the striking of the rocks and gravel against it when the buckets dump and it should be lined with an easily replaceable steel plate. A good way is to suspend a ½-in. plate from the top by means of a few bolts, leaving the bottom entirely free.

SCREEN

74 Two types of screen are in common use, the revolving and the shaking. The first dredges used the revolving screen, then the shaking type was introduced at Oroville and for a time most of the dredges were built in this way. On the Yuba it was found that these did not break up the clay in a satisfactory manner and revolving screens were introduced. This type proved better for this district, and since then many have gone back to the revolving screen even in other districts. The object of a screen is not only to size the material passing through it, but also to disintegrate and wash it. This the revolving screen does more thoroughly than the shaking screen where clay or cemented gravel is present. The shaking screen gives a more even distribution of the fines, requires less power to operate, and its plates last longer and are more easily replaced. For easily washed material the writer prefers the shaking screen, but for gravel hard to wash, the revolving type is decidedly preferable.

75 *Revolving Screen.* The design of the revolving screen (Fig. 20) has been well worked out and very little trouble is experienced with this part of the dredge. Most of the delays in its operation are caused by the necessary replacement of worn plates. These plates should be put in so as to be readily removable and not too close a fit made; rather leave considerable space between them, since if this is not done they will spread and be hard to take out. Bolts with countersunk heads on the inside and split washers under the nuts on the outside are used in preference to rivets. They hold as well and can be removed more readily.

76 The screens are built up in sections usually with cast-steel ribs at intervals to retard the movement of the gravel. These steel ribs also serve as a framework, to which are fastened the perforated plates. The size of perforations depends on the conditions and varies from ⅜ to ¾ in.

77 Around each end of the screen is a steel ring or tire preferably made in halves so as to be readily removable. Two drive rollers on which the ring at the upper end of the screen rests are used on most of the older dredges, but a single drive roller under the center with

two side guide rollers is coming into use on the newer boats. The rollers driving the screen are themselves driven by beveled gears and pinions. Owing to the difficulty of keeping grit out of these gears they are made of manganese steel, with a soft iron core cast in the hub. Rollers are used at the lower end to support the screen and a pair of manganese steel thrust rollers are provided to keep the screen from moving toward the stern. The bevel gears are driven from a countershaft above the screen, the driving motor also being located there. The countershaft is used in order to reduce the speed properly, so that excessively large gears or pulleys need not be used.

78 At the lower end of the screen is a hopper which directs the coarse gravel onto a belt conveyor. In order to prevent clogging at this point the hopper should be placed on a grade of not less than 30 deg. and should be raised on the side on which the revolving screen moves upward, the gravel being carried up on this side of the screen.

79 *Shaking Screen.* The shaking screen (Fig. 21) is built with channel iron sides. The bottom is formed of angle iron cross-pieces, to which are bolted the perforated plates. The angles should extend beyond the sides so that the top of the channels may be braced from their ends. The screens are built in pairs and mounted tandem so as to balance each other, one moving forward when the other moves aft. They are suspended from the timbers above, some using adjustable, others non-adjustable rods. The upper and lower bearings for these rods should be made of ample size, since they wear surprisingly fast considering the small movement. The lower end of the after screen is constructed so as to direct the coarse material to the short discharge shoot onto the conveyor. This part of the screen is not perforated. The size of screen perforations depends on conditions and as in the revolving screen varies from $\frac{3}{8}$ to $\frac{3}{4}$ in.

80 The shaking screens are driven by means of four connecting rods, operated either by cranks or eccentrics on a thwartship shaft. The author is inclined to favor the eccentric drive owing to its larger wearing surface. If eccentrics are used they should have long hubs and double keyways, otherwise they are likely to work loose under the constant pounding. The drive shaft may be placed at any convenient point, but a central location between the two screens is perhaps the best. The bearings for this shaft are placed outside the screens, and as near as possible to the cranks or eccentrics, and should be long enough so that they will not beat out under the constant hammering. Buffer springs are sometimes used to take up the strain at the end of each stroke, but this practice is not common. The connecting rods are

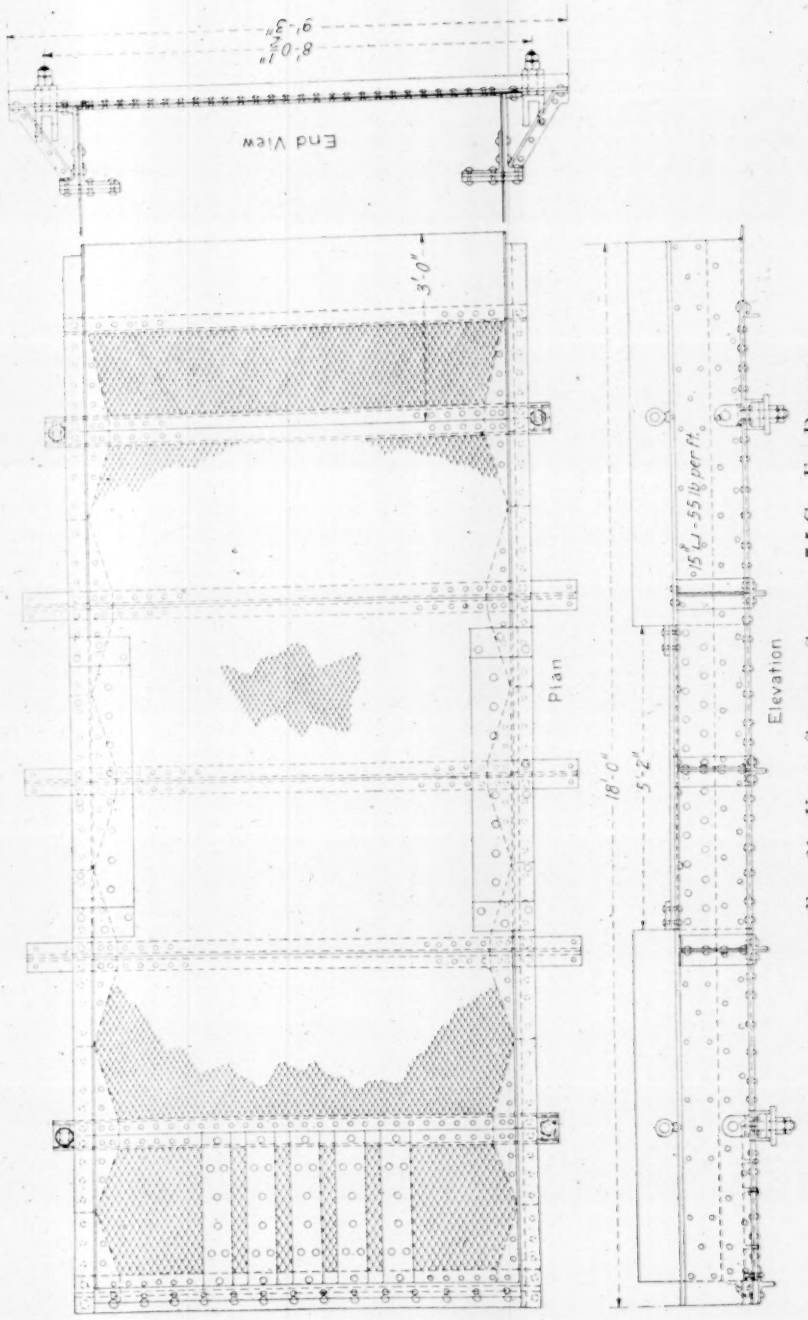


FIG. 21 UPPER SHAKING SCREEN, 7.5-CU. FT. DREDGE

usually made adjustable and any of the standard methods of fastening rod ends may be employed.

81 *Sprays.* The ordinary form of spray for washing the gravel as it passes down the screen is a perforated pipe or pipes. The water from the high-pressure pump is used, and the holes vary from $\frac{1}{4}$ to $\frac{3}{4}$ in. diameter. These holes are so placed as to direct the water where most needed. The spray pipes should have quick-opening valves at their lower ends so that they may be flushed out when the holes become clogged with roots, grass and other small particles of matter. Where much clay has to be washed some of the dredges are using large-sized nozzles set in groups at each end of the screen. It is thought that the large streams break up the clay in a more satisfactory manner than the greater number of smaller-sized streams furnished by the perforated pipes.

STACKER

82 *Ladder.* The design of a stacker ladder is a simple matter, since the loads and conditions are fairly constant. The lattice girder is the most common type and the strains in each member may be readily ascertained by graphic methods. The author uses a factor of safety of eight for this ladder. The lower end is supported near the stern of the boat and should be so designed as to allow the ladder to move up and down and also to allow the ladder to swing from side to side. The upper portion is supported by a bridle near the end and one near the middle. The ends of a wire rope are made fast to these bridles, its middle portion passing over a sheave which is in turn attached to the stacker ladder hoisting blocks. Instead of a single loop arrangement I prefer to fasten one end of the rope to the ladder hoist block, from thence passing it around a sheave at the upper bridle, then over another at the ladder hoist block, fastening the end to the middle bridle. This puts nearly double the supporting power at the upper end and prevents the buckling upward of the ladder when the dredge is surging badly.

83 *Conveyor Drive.* The common method of driving the belt conveyor is by means of a motor set on supports above the ladder 15 ft. or so back from its upper end. The motor is belted to a pulley shaft mounted near the end of the ladder, onto which is keyed a pinion with cut teeth which meshes into a cut gear on the conveyor belt pulley shaft. The author recommends a large size drive pulley covered with rubber belting. In order to make the belt contact greater an idler is placed near this pulley which brings the belt up and in con-

tact with it for a large proportion of its circumference. By mounting the motor inside the ladder near the upper end a direct gear drive may be used, but this method has not been found as satisfactory as the belt drive described above. A silent chain drive is now being used a great deal. It allows the motor to be set nearer the drive and is said to work very well.

84 *Conveyor Belt.* Rubber conveyor belts are used and should have an extra thickness of rubber on the wearing side. The most important point is to select rubber of the very best quality.

DISTRIBUTOR

85 Below the screen is a hopper or distributor which is used to distribute the screened material onto the gold-saving tables. Here the ingenuity of the engineer may have full sway. The main object is to get the screened gravel evenly distributed on the gold-saving tables, and the way that it is accomplished makes very little difference. The distributor should be designed so that a certain amount of adjustment will be allowed to meet varying conditions. Where a shaking screen is used the problem is simpler than for a revolving screen.

GOLD-SAVING TABLES

86 This part of the dredge belongs more to the province of the mining than the mechanical engineer, so it will not be discussed at any length in this paper. Suffice it to say that the gold-saving tables should be specially designed to meet each different set of conditions. They occupy the greater portion of the dredge back of the main gantry and may be built either of plate steel or wood. The author prefers steel. The tables are covered with riffles made of angle iron or strips of wood covered with strap iron. The fine material is discharged either at the stern or some distance aft according to conditions.

LADDER HOIST

87 The ladder hoist (Fig. 22) is now usually driven by the digging motor from a countershaft located on the main deck. In a few cases it is driven from the bucket-drive pulley shaft on the main gantry. Owing to the load it has to carry it is made very heavy. A structural or cast-steel base is provided, on which are mounted the bearings. Single gearing is used and ordinarily the teeth are not cut. A friction clutch is provided on the pulley shaft to connect this part of the dredge machinery. From the standpoint of rope efficiency the drums

should be made much larger than is common practice, but excessively large drums are not warranted, unless the sheaves through which the same rope passes are made of corresponding dimensions. This entails

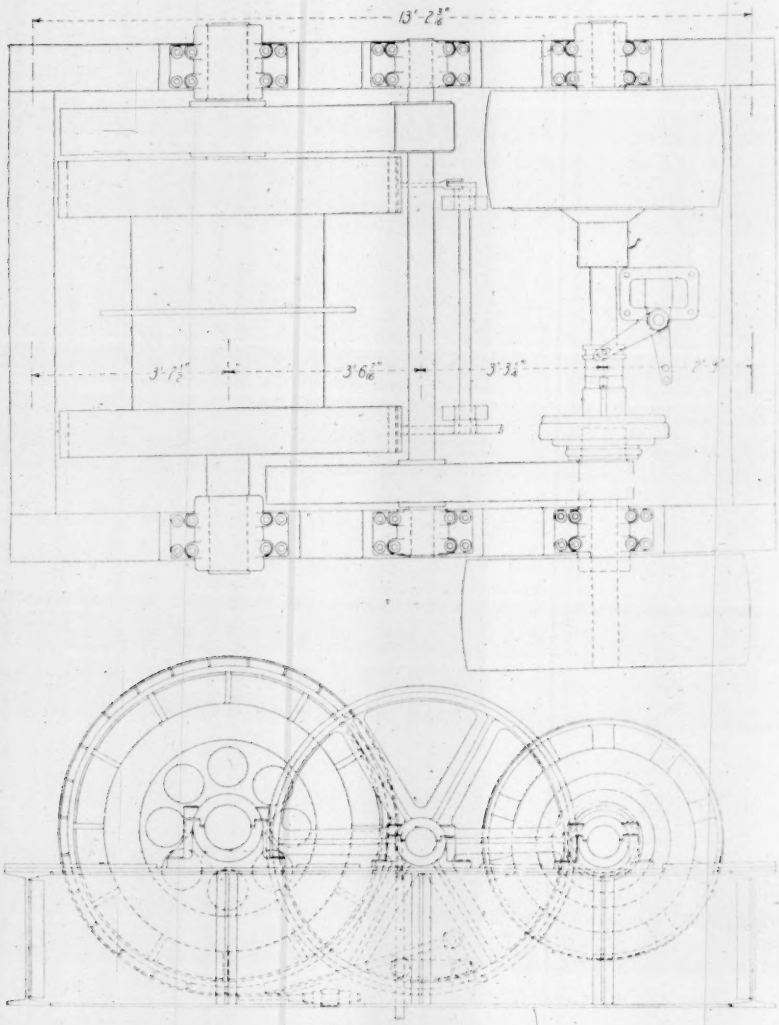


FIG. 22 LADDER HOIST FOR 9-CU. FT. DREDGE

increased construction cost, but the author is inclined to the opinion that it would reduce operating costs sufficiently to warrant some increase in this direction beyond common practice.

88 The brakes on the ladder hoist must be adequate in capacity

and made so that there may be no possible danger of their giving way. If the brakes on the main winch should give way little damage is likely to occur, but if they fail on the ladder winch a broken digging ladder is the probable result, if nothing worse. The following arrangement has been found very satisfactory. Two band brakes are used, one being operated by hand from the pilot house, the fixed end of which instead of being fastened to the deck is attached to a lever arm, which through a cross shaft operates the other brake, thus giving almost unlimited power with very little effort.

89 On most ladder hoists the drum-shaft gear is keyed to the drum shaft and it is nearly always located very close to the drum. It is a much better plan to design the side of the drum in such a way that the gear rim can be bolted to it, thus doing away with the gear spokes, making a much stronger job and taking all driving strain off the drum shaft.

MAIN WINCH

90 The main winch (Fig. 23) operates the swinging lines, both bow and stern, raises the spuds, and is usually fitted with one or two head-line drums for wrecking and repair work and to step up with when one spud has been broken. The winch is driven by a variable speed motor which together with the levers actuating the clutches is operated from the pilot house. Some form of friction clutch is commonly used on each drum. The writer has for some time been using an electro-magnet clutch on the pulley shaft of the winch similar to the one described for the bucket drive, and jaw clutches for each drum. The jaw clutches are extremely simple and require no attention, whereas a friction clutch is likely to get out of adjustment. Also each winchman wants his clutch set differently and unless great vigilance is exercised they will be constantly adjusting these clutches, which means lost time. The friction clutches on each drum are easier to operate, but with a little practice the jaw clutches give perfect satisfaction.

91 A good many dredges have a separate drum mounted on the after part of the dredge driven from the countershaft of the screen drive and is used to raise and lower the stack ladder. The drum is driven by a worm gear of such pitch that it will not run backwards from the weight of the ladder. Other dredges have an extra drum mounted with the main winch for this purpose. It is kept from reversing by a pall and ratchet. Considering the little use this drum gets its location and design seem to be of very little moment, but a worm drive and separate drum mounted aft are preferable.

92 *Swinging Lines.* On deep ground and for the larger dredges the bow swinging lines pass from the drum to a sheave or combination of sheaves located near the bow, then to a sheave on the shore, and from there to the lower end of the ladder. By this method of rigging very little side strain is put on the digging ladder or well hole. The disadvantage of this method is the necessity of a shore sheave and the consequent wear and time lost in moving and handling it. Where the ground is not too deep, a single line, which is simple to operate but which puts a large side-bending strain on the ladder and well hole is used. If the ladder and hull are built strong enough, this is not an objection. For the 13- and 16-ft. dredges a single line would have to be very large and would hardly be advisable.

93 *Brakes.* All winch drums are provided with band brakes operated from the pilot house. The connection between brakes and clutches and pilot house are made through bell cranks, rods, etc.

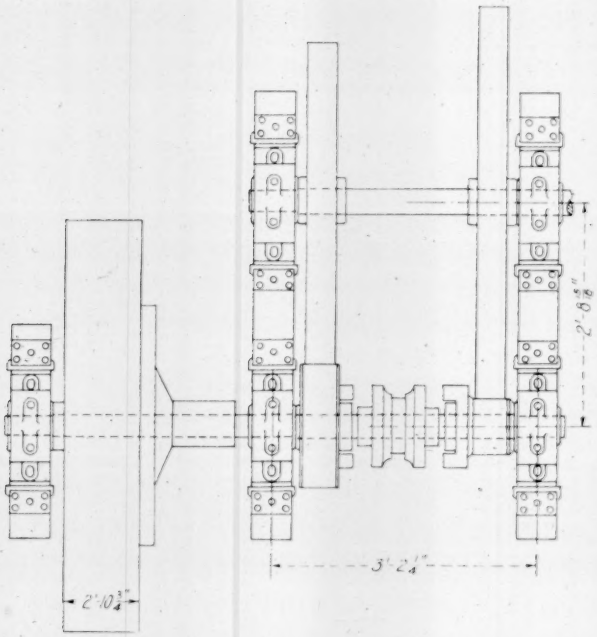


FIG. 23-a MAGNETIC CLUTCH AND DRIVE FOR MAIN WINCH, FIG. 23-b

Flexible wire rope running over sheaves is used in several instances for the brakes.

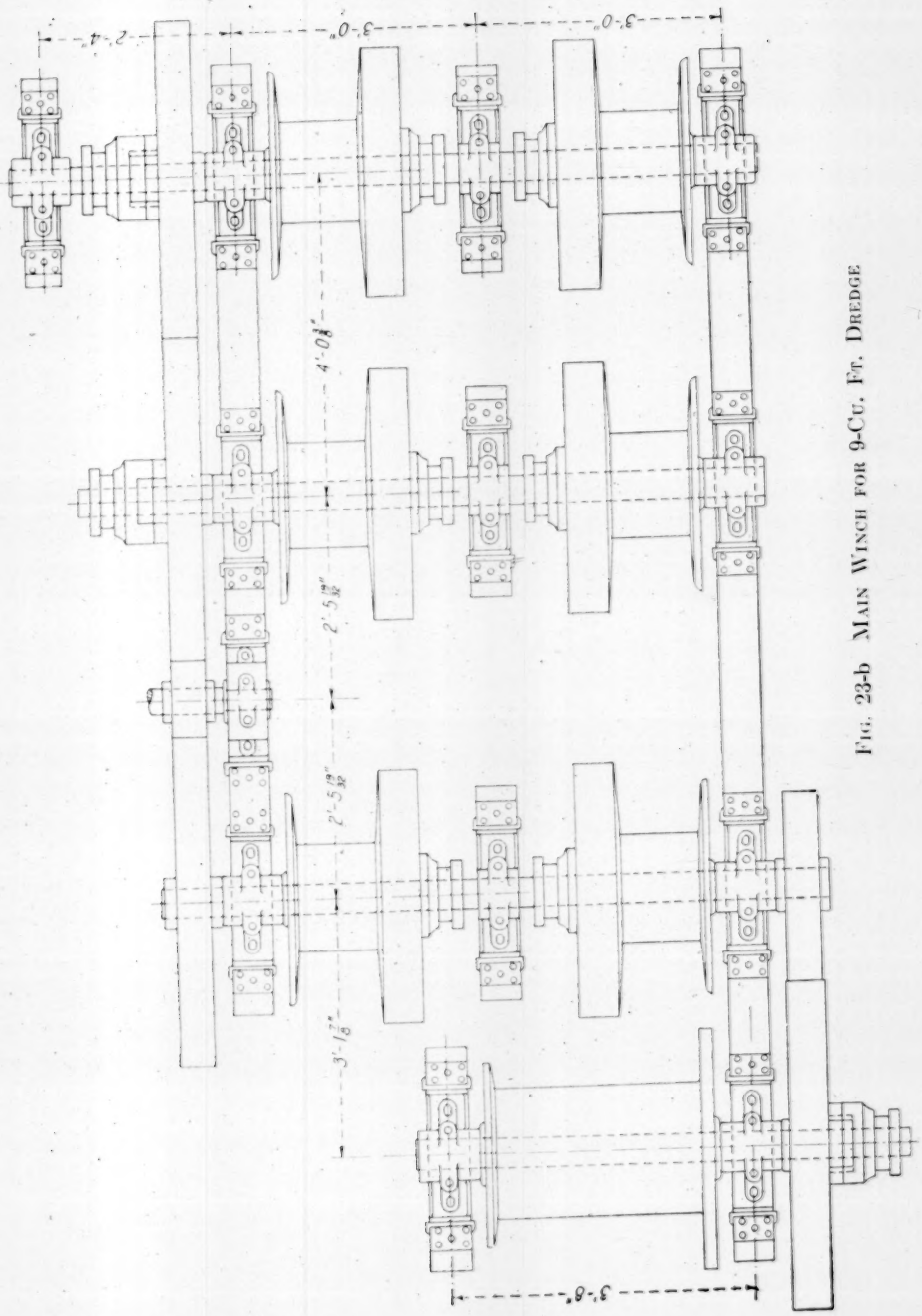


FIG. 23-b MAIN WINCH FOR 9-CU. FT. DREDGE

94 *Foundations.* Cast iron, cast steel, structural steel and wood are used for foundations. The author has been using wood in his recent designs on account of its producing less noise and vibration than metal foundations.

WIRE ROPE

95 Wire rope is used on the dredges for hoisting the digging and stack ladder, spuds, for all the swinging lines and for guying the gantries. On account of the small diameter sheaves, the necessity of the ropes being under water part of the time and the general hard service they get, their life is short as compared with ropes used for most other purposes.

96 Some dredge constructors use the regular open and closed sockets for their guy rope ends, and others think this practice is dangerous and use clips. The writer has never had any trouble with standard rope sockets if properly attached to the rope end. This may be done by first spreading the wire strands, treating them with dilute hydrochloric acid, inserting in the socket with the strands spread broom fashion, and then filling with melted zinc. The ends of the wires should not be turned over, but left straight. If turned over some are likely to get more strain than others, and to break and cause trouble.

97 Different makes and types of wire rope are used. The author's experience points to standard 6/19 plow steel hoisting rope for ladder, spud and side lines and 6/7 crucible cast steel standing rope for the guys. Hot Stockholm tar makes a good dressing, also most of the rope greases to be had on the market. Whatever dressing is employed it should be used freely.

SHEAVES

98 The sheaves used for the various purposes are made of cast steel in preference to cast iron. They are bushed with bronze or cast iron and their hubs made as long as possible. When long hubs cannot well be used, as in multiple blocks, the pins should be made of large diameter, so that the pressure will not be too great on each unit of resisting surface.

PUMPS

99 Three centrifugal pumps are ordinarily used on the gold dredge. The high-pressure pump should be designed to have a working head of at least 60 ft.; more is better if the gravel is at all hard to wash. This pump furnishes the water for the spray pipes or jets in the screens and is direct-connected to its motor.

100 The low-pressure pump is likewise driven by a direct-connected motor and should have a head of 20 to 30 ft. This water is piped to the tables where needed, depending on the nature of the gravel and the arrangement of the gold-saving tables.

101 The priming or fire pump is usually a two-step pump with a working head of 100 ft. or more. This pump is used in case of fire, for washing down decks, in cleaning up, for pumping out the bilge, and for priming the larger pumps.

102 All pumps should be fitted with both foot and check valves Bypass piping equipped with the requisite valves should be put in to be used for priming. The foot valves are provided with a grating, and the suction should be still further protected by a wire netting box attached to the side of the dredge. If this box be made of ample dimensions, say, 8 ft. by 15 ft., a $\frac{3}{8}$-in. mesh will work well. If a smaller box be used, $\frac{1}{2}$- or $\frac{5}{8}$-in. screen will have to be used, otherwise it is liable to get stopped up.

MOTORS

103 Three-phase induction motors are used on all standard California dredges, variable-speed for the digging and winch motors and constant-speed for the other pieces of machinery. Lately a form of variable-speed motor has been used for the stacker and screen drive, which gives a better starting torque for getting these parts under way and is particularly useful when it is necessary to start under full or overload. Any of the standard types of motor give most excellent results if the proper sizes are provided.

TRANSFORMERS

104 It is customary to bring the current onto the dredge at 2200 or 4400 volts, then transform down on the dredge with three oil-cooled transformers to 440 volts. Many of the later boats are using 2200-volt motors, thus avoiding the use of transformers altogether. Beside the three main transformers, another small one reduces the current to 110 volts for lighting purpose. All of the transformers are mounted in steel or asbestos-lined compartments opening to the outside of the dredge. They have pipe connections which allow the oil to be drained off and conveyed overboard to a point some distance from the transformers. These precautions are necessary in order to secure the best insurance rates and are to be recommended as a safeguard against fire. Another precaution which might prove useful is to provide the transformer housings with plate steel doors hung from

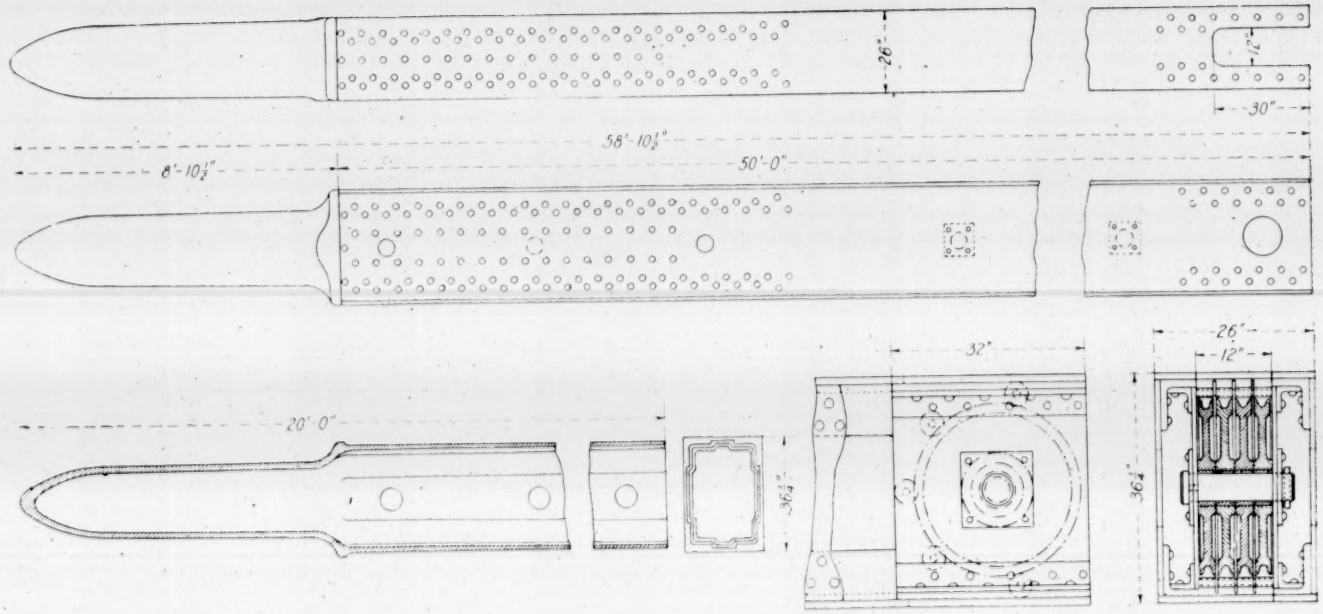


FIG. 24 SPUD FOR 5-CU. FT. DREDGE, SHOWING SECTIONS

the top with an automatic latch at the bottom and so constructed as to be readily dropped and latched in case of fire.

WIRING

105 The electric current is brought on board the dredge on floats in a three-conductor armored cable at a point near the stern, this part of the hull having the least movement. An oil switch is located near this point and is often so arranged that it may be instantly thrown out from the pilot house. From this switch the current is conveyed to the transformers in case low-voltage motors are used and from there the secondary is distributed to the different switchboards. The latest boats use conduits and condulets for all the wiring. If open wiring is used the conductors should not be stretched too tightly and they should all be stranded. The single wires are likely to break under the severe vibration of digging and the movement of the hull and superstructures. Stranded conductors even with the conduit wiring are recommended by the writer. For the motors located on the lower deck their wiring connections can be made through the hold. For the other motors the best method must be determined according to the conditions.

SPUDS

106 The early California dredges used a head line to dig on and even after the spud system was introduced it was a mooted question which was the better. The standard California type dredge uses the spud system (Fig. 24) exclusively, the head line being abandoned except in a few cases where the nature of the gravel or other conditions are unusual. At first, wooden spuds were used, then one wooden and one steel, the wooden spud being used to step up with and the steel one to dig on. Now both spuds are often made of steel. This plan is by far the best, since if one breaks the other may be used to dig on, using the head line to step up with. If only one steel spud is provided and a break occurs, the dredge must be shut down at once and the broken spud taken out, repaired and put in again before any digging can be done. Several forms of spud are used, and their make-up can best be understood by referring to Fig. 24, which shows views and cross-sections of a common design. An extra cover plate riveted to the tension side of that portion of the spud having the greatest strain adds strength with comparatively little increased weight.

107 The lower ends of the spuds are protected by cast-steel points, which extend far enough into the structural part to give ample

riveting area. The later designs are made longer than the older ones (Fig. 22). At the bottom they are acorn-shaped and change gradually from a circular to a rectangular cross-section at the point where they enter the structural portion of the spud, and they often extend 8 or 10 ft. below this point.

108 Heavy car springs are often used in the spud frames to reduce the shocks of digging and those using them think that they lessen the danger of breaking very much. This is doubtless true, but the author is of the opinion that it also reduces the digging capacity of the dredge and that the best remedy for broken spuds is to build them stronger.

CRANES AND REPAIR DEVICES

109 The modern dredges are provided with traveling or boom cranes covering the heavy machinery so that repairs and replacements may be made expeditiously and without calling in more men than the regular dredge crew. A thwartship I-beam with traveler, or a swinging boom covering the forward portion of the dredge and bucket line, is used for changing buckets. Brackets are riveted to the ladder, into which chains are hooked to hold the bucket line after parting. It is well to have an I-beam or at least an eye or hook well secured above the ladder hoist, main winch, pumps, transformers and screen. Over the bucket line a thwartship traveling crane is most useful. It should be of sufficient capacity to lift the upper tumbler, shaft and gears and should extend far enough so that these parts may be lowered over the side of the dredge. If a hatchway is provided through the upper deck on each side of the bucket drive this crane may also prove useful for handling parts on the lower deck.

HULL

110 The hull is built of Douglas fir and is proportioned to accommodate properly the machinery which it supports. Before designing the hull the general dimensions and weights of all machinery should be determined and the hull built accordingly. The depth largely depends on the weight to be supported, the beam on the gold-saving area desired, the length and width of well on the dimensions of the digging ladder, and the digging ladder in turn on the depth of gravel to be dug. The distance from well to stern is determined by the length of screen and the general distribution of machinery fore and aft. The hull rides on an even keel when digging at an average depth and should have a free board of not less than one-fourth its total depth.

111 The cross-section is rectangular, with the deck slightly crowned. Thwartship frames are used, the bottom planking being spiked to the bottom thwartship timbers, the decking to the upper timbers or deck beams, and the side planking bolted to the upright side posts. Besides these, two or more pairs of posts are put in each frame and to these the well planking and bulkheads, or fore-and-aft trusses are bolted. Clamps or fore-and-aft strengthening timbers are bolted inside the frames at the corners where the posts join the deck beams and bottom thwartship timbers. The size of these clamps depends on the size of the hull. Drift bolts are driven in the side planking and bulkheads, each bolt passing through $2\frac{1}{2}$ planks. They may be put in as thickly as desired, and add considerable strength to the planking and bulkheads. Four-inch deck and bottom and 6-in. side planking is commonly used. The stern is often planked with the same size as the sides, put edgeways instead of flat. At the point where the spuds are located a short bulkhead or extra bracing is often used. Just aft of the well and below the bucket drive, one or more vertical cross trusses are often used, likewise horizontal cross bracing is put in on top of the thwartship frame timbers along the compartments adjacent to the well and extending some distance aft from the well throat.

112 The bow is made narrower than the rest of the barge, the frames being shortened and the side planking steamed and bent around them. How much this portion of the hull should be narrowed down will depend on the width of cut it is desired to make, together with the distance the digging ladder projects beyond the bow. Using one spud as a center and drawing the outline of the hull, digging ladder and cut to scale, will graphically illustrate how much to bring in the bows so that the dredge may dig its way along any desired width of cut.

113 The above is a general description of the most common type of hull construction. The writer has been using what he considers a somewhat stronger construction, but one which is more expensive. Fore-and-aft stringers are laid below the thwartship timbers. The posts are bolted to both the fore-and-aft and the thwartship timbers, the bottom planking being laid athwartships. Clamps are used above the thwartship timbers the same as in the other form of construction, the difference being that there is an extra set of fore-and-aft timbers, to which the bottom planking is spiked, this planking running athwartships instead of fore and aft. This construction requires more lumber and labor, but is stronger, and by using thwart-

ship planking full lengths can be used, doing away with all butt joints. The tendency of dredge hulls is to sag along the fore-and-aft center line. Where fore-and-aft planking is used this movement of the hull tends to open up the seams, but where thwartship planking is used the strain is lengthways of the planking. When a hull is new these weaknesses do not show, but after years of service all such little things count, and since the life of a hull determines the life of a dredge, it pays in the long run to use every precaution possible to add to the stability and length of life of the hull. Solid fore-and-aft bulkheads are recommended, also the use of well plates made up of 6 in. by 14 in. planking (for the 7 to 9 cu. ft. dredge), laid horizontally below the deck beams and filling the roof of the first compartment on each side of the well for 30 ft. each way fore and aft from the throat of the well. These well plates are particularly useful in dredges having long well holes.

114 Another plan which will add strength at small cost is to put in diagonal wooden braces from the fore-and-aft timbers near the bottom of the intermediate bulkhead to the clamp near the top of the well hole. This will aid materially in preventing sag toward the well hole.

115 Still another form of strengthening construction being used, particularly on the larger boats, is an overhead thwartship truss with heavy hog rods running diagonally from the upper chord near the sides of the dredge to the bottom of the hull at the well, thus preventing sag at this point.

116 The bottom, sides, bow, stern and deck are all calked with oakum and the seams pitched or similarly treated. The inside is treated with carbolinium, Stockholm tar, painted or left without any treatment whatever. On the outside below the water line the planking may be painted or not as desired. Above the water line all timber work is thoroughly painted. Before putting timbers together all joints are painted with white lead, mineral paint, or coated with tar or carbolinium.

117 The deck beams are extended beyond the sides of the barge at the after portion of the dredge. Braces are put under these in order to support the housing uprights. This gives extra room for gold-saving area or passageways. Hatchways are provided for entrance into each compartment, also a system of ventilation should be installed, using either mechanically driven fans or natural ventilation induced by cowled ventilators. Proper ventilation will do much towards prolonging the life of the hull. Sea cocks are often installed

in order to sink the dredge in case of a fire which has gone beyond hope of being extinguished. If sea cocks are used they should be located near the bows where a hot fire in the housing will not prevent their being approached and quickly opened.

118 *Main Truss.* A large proportion of the California dredges have a truss along the sides of the well and extending from bow to stern gantry, with wooden chords, posts and braces and steel hog rods. This truss is made 15 to 25 ft. high, the upper chord often being used to support the upper tumbler bearings and the upper end of the digging ladder. In the earlier dredges there was a tendency for the hulls to go down at bow and stern and hump up near the throat of the well. This truss assists in preventing this condition, strengthens all the gantries and spud frame, and makes an excellent support for the housing, screens, etc. It should be designed as a truss supported at the middle with weights suspended at each end.

119 *Bow Gantry.* At the bow is a gantry (Fig. 25) which supports the forward end of the digging ladder, the details varying in the different dredges. A steel cap of box or double I-beam section on four posts is a common type. In some dredges the four posts are parallel; others have two posts parallel to the well sides and two sloping inward joining the first at the cap. Still others have the four posts parallel for half their length, the two outside ones being cut at this point, and from here diagonal braces run to the intersection of cap and inside posts. The whole gantry is inclined forward in order that it may have a more direct pull on the digging ladder. In a good many of the California dredges the gantry posts go through the deck and are stepped on heavy foundation timbers or castings near the bottom of the hull. A few dredges have rigid foundations for these posts on the deck, and at least one uses a socket casting into which the posts fit loosely.

120 This gantry has thwartship bracing and is guyed by ropes or tie rods to the main gantry, hull timbers or both. The top chord of the main truss runs to the middle portion of this gantry and is fastened to it by means of steel plates. It is also customary to run a brace diagonally from near the cap to the top of the first truss post. Some form of pin and plate clamp is provided for the bow gantry cap, to which is attached the ladder hoist blocks and gantry guys.

121 *Main Gantry.* The main gantry is a support for the bucket drive and the after end of the digging ladder. It has a very severe duty to perform and should be built in a most substantial manner. The author prefers to place a post under each bearing and use heavy

lattice bracing between the posts. Too many plates should not be used to tie the timbers together, and arrange matters if possible so that all bolts will be accessible. Where many plates are used it is very difficult to assemble the parts and in case of repairs the delay

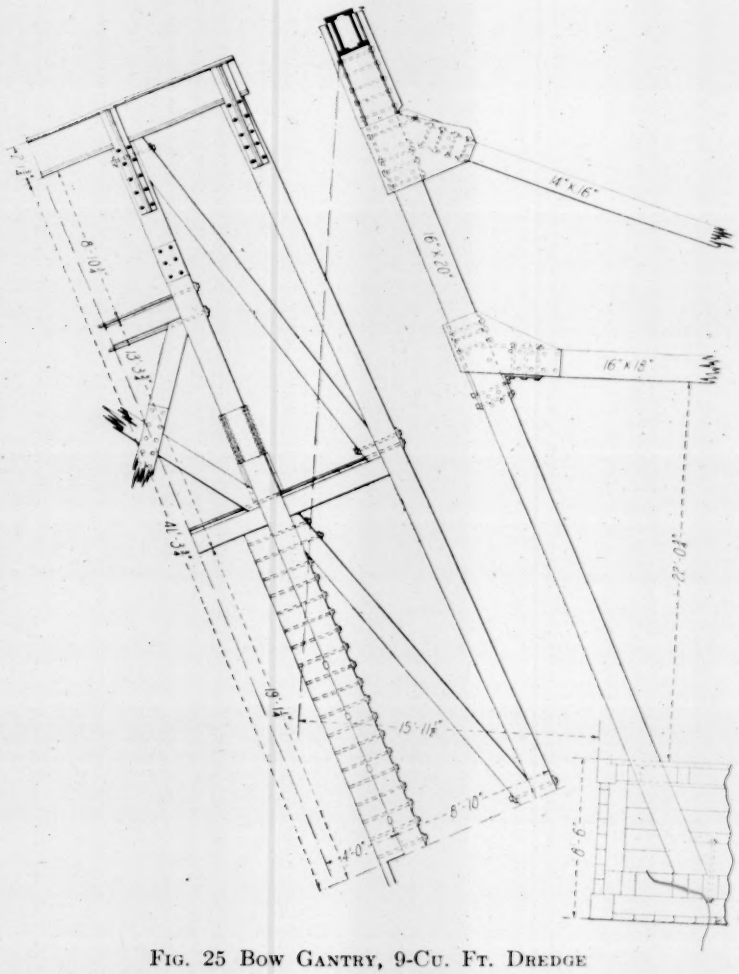


FIG. 25 BOW GANTRY, 9-CU. FT. DREDGE

is great in taking apart and re-assembling, owing to the difficulty of lining up the holes in the different plates.

122 *Stern Gantry.* The stern gantry (Fig. 26) supports the spuds and the stack ladder and must be made high enough to raise the

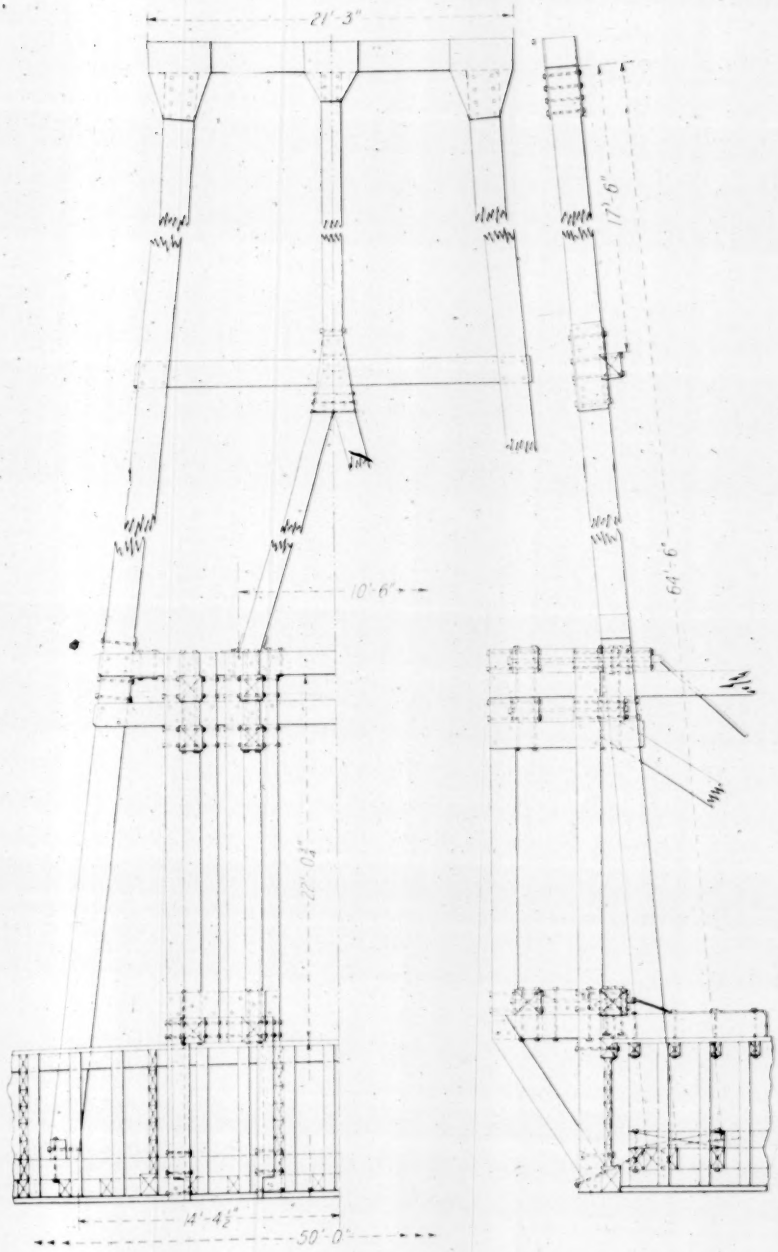


FIG. 26 STERN GANTRY, 9-CU. FT. DREDGE

spuds so that they will nearly clear the bottom of the hull. It is inclined slightly aft so that the spud hoist blocks will be vertically above the center of the spuds. The design is simple, two, three or four posts being used with a wooden or steel cap. Clamps are used for the spud hoist blocks, ladder support and guy ropes. The posts may rest on foundation timbers or castings located near the bottom of the hull or on the deck.

123 *Spud Housing.* At the stern of the dredge the two spuds are enclosed in a housing which allows them to slide up and down but holds them in a vertical position. At some point above the deck, usually at the elevation of the upper chord of the main truss, heavy timbers are framed around the spud, with tie rod connections to the hull or main gantry. This framing together with the tie-rod connections resists the tendency of the upper part of the spud to move backward as the buckets engage the gravel and thrust the hull toward the rear. This clamp must be made strong enough to withstand the full digging power of the dredge, increased to some extent by the slight fore-and-aft movement of the hull and the consequent momentum of the full weight of hull and machinery. The writer does not approve of running the tie rods to the main gantry, since no matter how strong this part is made the constant pulling and jerking on one side will tend after a time to throw the bucket-line shafting out of line. The guy rods should be run into the hold or to some other part which does not support shafting.

124 Many different spud-housing designs are used, but the present tendency seems to be to use heavy castings covering a large area of the stern to resist the forward pressure of the spud and a structural or cast-steel clamp 20 ft. or so above the deck. In order to keep the spud from moving backward heavy steel keepers are used which can quickly be removed when it is necessary to change spuds. The timber construction will work well enough, but it is likely to get loose in time, and it is difficult to design a wooden structure so that it will be at the same time strong and easily removable. Whatever form of spud housing is used it should be so enclosed that small rocks can not lodge where they will be caught between the spud and its guide.

125 *Housing.* The after portion of the dredge is housed in, this housing covering the gold-saving tables, pumps, main winch, ladder hoist and transformers. The after central portion is extended up and covers the screen and screen drive. A covering should also be provided for the bucket drive, but this is not essential. The roof of the housing is made with a slight crown or slope to the sides and

covered with painted canvas, ship fashion. The roof joists should be made heavy enough to support a considerable weight, since in repairing it is often convenient to lay down one of the big gears or the like, and for this reason the flooring should be at least $1\frac{1}{4}$ in. thick, or made of two thicknesses of planking. The stern and sides along the gold-saving tables are not usually boarded up. It is thought that an unobstructed view lessens the danger of amalgam stealing.

126 *Pilot House.* The pilot house from which the digging motor and winches are handled is usually located on the starboard side of the upper deck above the main winch and a little forward of the bucket drive. This is the common location, but the pilot house on the central line of the dredge above the bucket line and just forward of the bucket drive seems preferable. The only objection to this location is that the lever connections to the winches are a little more complicated. This location gives an excellent view in all directions around the dredge and down the bucket line. By proper arrangement of gratings the main winch may be seen. The bucket drive, stacker and spuds are also within sight of the winchman. In the writer's opinion there is no question as to the proper location for the pilot house, and all his dredge designs have their pilot houses well up in the center of the boat.

DIMENSIONS

127 In Table 1 the more important dimensions and some other data of several California gold dredges built at different times, of different sizes, and for different conditions are given. Most of the data were kindly furnished by the makers, and will serve to give a general idea of typical dimensions. In checking over the figures furnished several discrepancies were noted, but on the whole the figures are no doubt approximately correct.

CONSTRUCTION COSTS

128 The construction costs vary greatly, of course, according to the quality of workmanship and material desired, facilities of the builders in the way of properly equipped shops and foundries, and the geographical situation of the ground on which the dredge is to be built. The following may be considered a fair average of many cost records in California, for rough estimating purposes:

129 For all the machinery, including motors, pumps, wire rope, electric supplies, etc., $12\frac{1}{2}$ cents per lb., delivered on the ground; for installing, $4\frac{1}{2}$ cents; or a total of 17 cents per lb. for the machinery installed. The lumber costs about $30 per 1000 ft. at Portland, and

this together with freight, bolts, nails, hog rods, steel plates, oakum, paint, etc., will make the hull material cost about $50 per 1000 ft. of lumber used; labor of building costs about $40 per 1000 ft., giving $90 per 1000 ft. as a total cost of the completed hull. In addition the direct costs chargeable to hull or machinery, insurance, traveling, superintendence, camp equipment, temporary buildings, shop, derrick, office expense, design, etc., amount to between $10,000 and $25,000, depending on whether new designs have to be gotten out, what construction tools and equipment are on hand, the size of the dredge, and location of the ground on which it is to be built. For a a complete dredge, $180 per ton, on a basis of its displacement, is a fair average.

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES

Name of maker	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
Date built	September 1897	1907		1910	1905	1911
Capacity bucket, cu.ft.	3.2	5	5	5.6	7.5	9
Depth to dig below water line, ft.	20	38	35	30	65	65
Depth to dig above water line, ft.	8	10	12	10	10	10
Nature of digging	Loose	Medium hard	Hard	Medium	Medium	Medium
Hull:						
Displacement with all machinery	3 ft. 6 in.	5 ft. 9 in.	5 ft. 8 in.	5 ft. 3 in.	6 ft. 6 in.	6 ft. 2 in.
Length, ft.	70	102	106½	88	115	115
Depth, ft.	5	7 ft. 9 in.	7 ft. 10½ in.	7 ft. 6 in.	8	8½
Width, ft.	29	36	40	36	40	50
Overhang on each side, ft.	0	2	....	0	2½	4
Length of well, ft.	42	56 (approx.)	51 ft. 1 in.	48	63	63
Free board, ft.	1½ (approx.)	2 ft. 3 in.	2 ft. 2½ in.	2 ft. 3 in.	1½	2 ft. 4 in.
Side planking, ft.	3	4	4	4	4	6
Bottom planking, ft.	3	4	4	4	4	4
Height of bow gantry above deck, ft.	22	36	34 ft. 6 in.	34	39 ft. 6½ in.	37
Height of stern gantry above deck, ft.	24	50	61 ft. 8 in.	36	58 ft. 9 in.	58
Height of main gantry above deck, ft.	16	23	21 ft. 8 in.	21 ft. 6 in.	20 ft. 2 in.	22
Total amount of lumber in hull, gantries, housing, etc., ft.	75,000	136,000	263,600	156,000	300,000	400,000
Digging ladder:						
Weight, lb.	14,000 (approx.)	48,400 (approx.)	53,575	33,000	79,000	94,400
Distance between centers, ft.	40	87	84½	71 ft. 6 in.	114 ft. 1 in.	115
Extension beyond bow, ft.	12 (approx.)	25½	26	18	38	43
Width, ft.	3	4½	5 ft. 8 in.	3 ft. 8 in.	4 ft. 7½ in.	4 ft. 7½ in.
Maximum depth girder, ft.	24	7	5 ft. 6 in.	54	8	8
How upper end is supported	Fixed hanger - shaft with telescope bearing	Concentric	Steel pin, 8 in.	Fixed hanger - shaft with telescope bearing	Concentric	Concentric
General design	Plate girder	Plate girder	Plate girder	Plate girder	Lattice girder	Lattice girder
Stacker:	Bucket elevator	Belt conveyor	Belt conveyor	Bucket elevator	Belt conveyor	Belt conveyor
Length ladder, ft.	42	90	106	72	136	139
Width ladder, ft.	3 ft. 2 in.	4	3 ft. 8¼ in.	3 ft. 8 in.	3 ft. 8 in.	4 ft. 4 in.
Depth ladder, in.	15	5	4	15	8	7
Weight ladder, lb.	12,000	18,000	15,000	19,365	22,500	24,000

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—*Continued*

Name of maker	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
General design	2 I-beams	Lattice girder	Lattice girder	2 I-beams	Lattice girder	Lattice girder
Width belt, in.	Buckets 15 in. pitch	30	30	Buckets 15 in. pitch	32	32
Speed belt	48 buckets per min.	350 ft. per min.	Variable	54 buckets per min.	350 ft. per min.	400 ft. per min.
Horsepower motor	Driven by main engine	15	20	10	25	35
Maximum height of discharge, ft.	20	30	35	30	50	54
Bucket line:	Open connected	Close connected	Close connected	Open connected	Close connected	Close connected
Number in line	30	82	93	40	93	84
Maximum speed of buckets per min.	16	22	22	14	19	50
Weight bucket complete, base, hood, lip, pin, bushing, lb.	570	1300	1490	1200	2187	2268
Pitch of bucket, in.	24	28 5/16	24 1/2	25	32 1/4	36
Number front eyes	2	2	2	2	3	2
Number back eyes	2	1	1	2	2	1
Length back eye, in.	4	16	19 3/8	6	8 1/8	16 3/4
Width front eye, in.	4	3 5/8	6 1/2	6	3 3/4	4
Diameter pin, in.	1 7/8	4 1/8	4 1/2	2 3/4	4 5/8	5 1/4
Length pin, in.	10 1/2	23 1/4	30 7/8	12 1/2	2 ft. 3 5/16 in.	24 7/8
Ladder roller:						
Number on ladder	6	12	19	10	17	17
Diameter in.	8	14	12	12	14	14
Diameter, shaft in roller, in.	2	3 15/16	3 7/16	3 1/2	4 1/2	5 7/16
Diameter shaft in bearing, in.	2	3 3/8	3 7/16	3 1/2	3 15/16	5
Length shaft in bearing, in.	6	11 1/8	....	8	12	11 7/8
Material used	White iron	High carbon and manganese steel	Semi steel	Manganese steel	High-carbon steel	Chrome steel
Weight roller and shaft, lb.	280	1030	1008	778	1160	1140
Upper tumbler:						
Diameter shaft in tumbler, in.	8	14	14	11	17	17
Diameter shaft in bearing, in.	8	12	12	11	14	14
Span between bearing	4 ft. 6 in.	6 ft. 6 in.	....	5 ft. 2 in.	9 ft. 3 3/16 in.	9 ft. 3 3/16 in.
Weight complete with shaft and plates, lb.	3010	11,200	10,185	6,220	18,750	21,310
Lower tumbler:						
Diameter shaft in tumbler, in.	6	10 1/4	11 3/4	10	11 1/4	12
Diameter shaft in bearings, in.	6	9	7 3/4	10	10	10

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—*Continued*

Name of maker	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
Length of shaft,	4 ft. 2 in.	4 ft. 9¼ in.	6 ft. 5⅞ in.	5 ft. 3 in.	4 ft. 11¾ in.	5 ft. 3½ in.
Maximum height side flanges at corners above wearing plates, in.	7¼	6⅝	9½	11	12½	11¼
Thickness side flanges at top and bottom, in.	1–2¼	1⅝–3⅛	2¼–3	1½–3¼	1–1⅜	2
Inside flare of side flanges	....	2 in. in 12 in.	Curved	....	⅜ in. in 12 in.	Straight
Weight complete with shaft and plates, lb.	4,176	10,000	8,355	7,746	11,340	12,758
Ladder hoist:						
Total weight, lb.	Part of main winch	14,800	20,000	5,660	....	39,000
Diameter of drum, in		18	30	18	20	3 ft. 6 in.
Diameter of drum shaft, in		6½	6⅞	....	8	9½
Speed of drum, r.p.m.		19.2	9.6	....	12.8	7.6
Maximum speed ladder line, ft. per min.	....	....	....	10	67	100
Horsepower required to raise ladder at above speed	....	....	....	35	....	120
Diameter of sheaves, in.	....	30	30	36	30	42
Main winch:						
Total weight, lb.	10,385	23,700	32,500	24,200	....	23,275
Horsepower of motor	12	20	35	25	25	25
Diameter of drums, in.	15	16	18	18 and 24	18	16
Diameter of drum-shaft, in.	3½	4⅞	4⅜	5⅛ and 7	4⅜	4⅜
Maximum number r.p.m. drum	....	14	20.5	2	9	16
Minimum number r.p.m. drum	....	2.6	....	....	2.5	4.7
Maximum speed of side swing at end of digging ladder, ft.	....	62	....	....	17.5	38
Minimum speed of side swing at end of digging ladder, ft.	....	11.5	....	....	5	11
Speed of raising spud, ft.	....	10	....	....	4.5	9.5
Horsepower necessary to raise spud at above speed.	....	10	....	....	6	15

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—*Continued*

Name of maker	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
Bucket drive:						
Horsepower digging motor	50 h.p. steam	100	100	100	150	200
Diameter and face engine pulley, in.	66	96 x 24	86 x 21	96	96 x 24	96 x 25½
Pitch diameter and face pulley shaft pin	....	10.82 in. P.D. x 7 in. F.	11.52 in. P.D. x 6½ in. F.	....	11.936 in. 8 in. F.	11.936 in.- 8¼ in. F.
Speed pulley shaft, r.p.m.	....	160	143	156	113	115
Speed intermediate shaft, r.p.m.	....	44	22.5	56	21	21
Pitch diameter and face intermediate shaft gears	....	39.67 in. x 7 in.	74.49 in. x 6 in.	47¾ in. x 6 in.	62.87 in.- 8 in. F.	62.87 in.- 10 in. F.
Pitch diameter and face intermediate shaft pinions	....	13.5 in. x 8 in.	14.32 in. x 10 in.	....	20.136 in.- 10 in. F.	20.136 in.- 10 in. F.
Speed tumbler shaft, r.p.m.	8	4½	4.5	7	3.1	3
Height tumbler shaft above deck,	....	23 ft.	25 ft. 2½in.	....	22 ft.5¼ in.	24 ft. 6 in.
Pitch diameter and face tumbler shaft gears (bull wheels).	....	9 ft. 11 in.- 8 in. F.	71.62 in.- 9 in. F.	....	143.55 in.- 10 in. F.	143.55 in.- 10 in. F.
Weight bull wheels, lb.	2,100	5,500	3,110	3,200	8,985	10,200
Spuds:						
Total weight including all castings, lb.	....	20,400	43,800	....	32,000	36,000
Total length, ft.	....	50	58 ft. 10 in.	....	60	61
Width, in.	....	24	26	....	24	24
Depth, in.	....	36	36	....	36	36⅞
Type of construction.	....	Box type	Box girder	....	Center web	Center web
Screen:						
Type	Revolving	Revolving	Shaking	Revolving	Revolving	Revolving
Total length, ft.	20	25½	32	24	30	36 ft. 3 in.
Width or diameter	3 ft. 6 in. diameter	6 ft.	7 ft. 4 in. upper 7 ft. 11 in. lower	4 ft. 6 in. diameter	6 ft. diameter	6 ft. diameter
Size of holes	⅜ in. dia.	⅜in.to½in.	⅜ in.	⅜ in. dia.	½in.to⅝in.	½ in.
Grade	1½ in. in 12 in.	2 in. in 12 in.	1⅞ in. in 12 in.	1½ in. in 12 in.	2 in. in 12 in.	2 in. in 12 in.
Horsepower of motor	Driven by main engine	20	20	15	35	50

TABLE 1 GENERAL DATA ON CALIFORNIA GOLD DREDGES—Continued

Name of maker.....	Risdon	Bucyrus	Link Belt	Risdon	Marion	Marysville Dred'g Co.
Number r.p.m. or strokes per min...	12	7	....	9	7	8
Gold-saving tables:						
Total area, sq. ft....	160	800	....	370	1600	2400
Grade.....	1¾ in. in 12 in.	1⅞ in. in 12 in.	1¼ in. in 12 in.	1¾ in. in 12 in.	1⅛ in. in 12 in.	1⅜ in. in 12 in.
Wood or steel.....	Steel	Wood	Steel	Steel	Wood	Steel
Width of bays, in...	28	36	30	32	33	36
Dimension of riffles	Cocoa matting and expanded metal	1⅞ in. x 1¼ in. wood	1 in. x 1 in. x ⅛ in. L.	Cocoa matting and expanded metal	1¼ in. x 1¼ x ⅛ in. L.	1¼ in. x 1¼ x ⅛ in. L.
High-pressure pump:						
Horsepower of motor	....	15	100	30	50	100
Head, ft.....	18	65	70	31	50	68
Quantity of water, gal. per min....	3000	500	2800	2400	....	3200
Size suction pipe, in..	....	4	12	14	10	12
Size discharge pipe, in.....	....		10	9	8	10
Make of pump.....	Risdon	Yuba Con. Co.	Worthington	Risdon	Worthington	Abner Doble
Low-pressure pump:						
Horsepower of motor	No low-pressure pump	50	No low-pressure pump	....	25	35
Head, ft.....		60		18.6	25	25
Quantity of water, gal. per min.....	....	2000	....	2200	....	3800
Size suction pipe, in..	....	8	....	....	10	12
Size discharge pipe, in.....	....			9	8	12
Make of pump.....	....	Yuba Con. Co.	....	Risdon	Worthington	Abner Doble
Fire pump:						
Horsepower of motor	....	Use high pressure	5	5	10	10
Head, ft.....	....		65	40	90	100
Quantity of water, gal. per min.....	....		150	....	....	....
Size suction pipe, in..	....		3	3	4	2½
Size discharge pipe, in.....	....		2½	2½	3	2
Make of pump.....	....		Worthington	Krogh.	Worthington	Abner Doble
Lines:						
Size ladder hoist, in..	⅝ in.—	⅞ to 1	1	⅞ in. diameter	1	1⅛
Size stacker hoist, in.		¾	¾	1½ in. diameter	¾	¾
Size spud hoist, in....	....	¾	¾	....	⅞ and ¾	¾
Size swinging lines, in.....	½—in. rope 300 ft.	¾ to ⅞	¾ to ⅞	⅞ and ¾	1 and ¾	1
Total weight machinery, tons.....	90	260	305	220	300	400

RESULTS OF TESTS ON THE DISCHARGE CAPACITY OF SAFETY VALVES

By E. F. Miller

ABSTRACT OF PAPER

The paper gives the results of a series of tests made on the discharge capacity of 3-in. and 3½-in. locomotive pop safety valves at varying lifts under 200 lb. boiler pressure, also results of tests on the discharge capacity of 3-in. and 3½-in. inspector valves of the flat-seated and bevel-seated types under 100 and 150 lb. gage pressure. In all of the tests the steam was condensed in a surface condenser and the condensation weighed. The tests were run from 30 minutes to 1 hour in duration so that errors of weighing were reduced to a minimum. In all of the tests the valve was set a definite distance from its seat and held rigidly, the spring of the valve having been removed.

Results of the tests are given both in the form of a plot and numerically.

RESULTS OF TESTS ON THE DISCHARGE CAPACITY OF SAFETY VALVES

By E. F. Miller, Boston, Mass.

Member of the Society

The tests quoted in this article (Tables 1 and 2) were made during the spring and summer of 1910 for the information of the Crosby Steam Gage & Valve Company, by whom permission has been given to present the contents of the report to the Society, believing that the results will be of some value to its members.

2 In all the tests the valves were without springs and were set a definite distance from their seats and held rigidly during the tests. They are shown in Figs. 1, 2 and 3 as modified for these tests. As a result of the tests the locomotive valve is now made as shown in Fig. 4. In order to avoid unequal expansion, the metal of the valve body and the metal of the spindle were made the same in the case of the locomotive valves; difference of expansion was similarly obviated as far as possible in the inspector valves. The spring was replaced by a spindle attached to the valve. This spindle extended through the top of the valve and carried at its free end a pointer which traveled over a graduated cylinder. The lower part of the spindle was threaded with a 20 thread. The graduated cylinder at the top was divided into five parts, each representing 0.01 in. A motion of the pointer of approximately 1 in. corresponded to a 0.01 in. lift of the valve from its seat. Tests on the 3-in. valves were run with lifts set at 0.10,0.08, 0.05 and 0.02. Tests on the $3\frac{1}{2}$-in. valves were run with the same lifts at 100 and 150 lb. pressure, but the capacity of the condenser prevented running the $3\frac{1}{2}$-in. valves with 0.10 in. lifts at 200 lb. pressure.

3 The valves were connected with the boilers through a line of 5-in. pipe, into which two Babcock & Wilcox boilers of 500 boiler h.p. discharged. The 5-in. pipe connected with a 10 ft. length of 10-in.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

pipe on the end of which was a blank flange 2 in. thick. To this flange the safety valve under test was bolted (see Fig. 5). This flange was bored on the bottom with a hole considerably larger than the inlet to the valve. The entrance edge of this hole was rounded with a curve of 1 in. radius. On the other side of this blank flange and enclosing the safety valve was a 10-in. flanged tee. The outlet

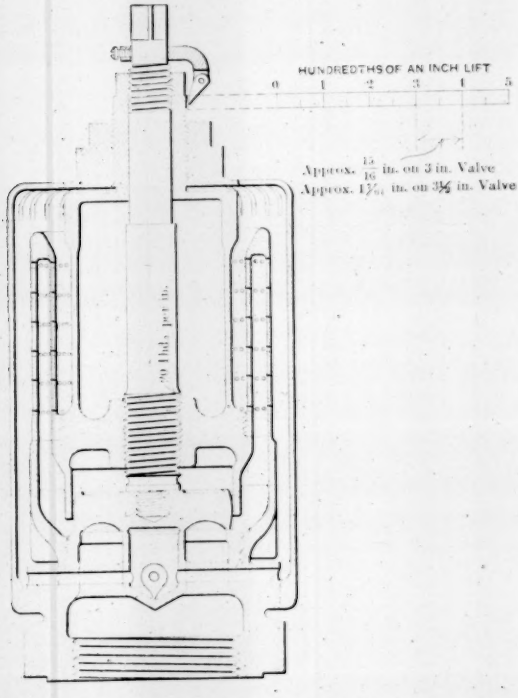


FIG. 1 CROSBY MUFFLED LOCOMOTIVE POP SAFETY VALVE

of this tee led to the condenser. The end of the tee on the straight run was covered with a blank flange. Between tests this blank flange on the end of the tee was removed and the setting of the valve changed. After each test the valve was examined to see if it had moved from its previous setting. The steam passing through the valve was condensed in a surface condenser vented to the air through a $2\frac{1}{2}$-in. pipe. The condenser was tested immediately before and after each test and was found to be absolutely tight. The condensed steam was pumped to two weighing tanks, each of about 1500 lb. capacity.

4 The steam was led from the boiler through a small separator

in the boiler room, then through a second separator located about 20 ft. from the safety valve. A calorimeter was attached to the 10-in. pipe near the safety valve and the quality of the steam determined. In all of the tests the steam was practically dry.

5 To see whether or not there was any pressure in the muffler or in the outer valve casing, a copper pipe was connected and led through a stuffing box outside of the 10-in. tee and a low-pressure

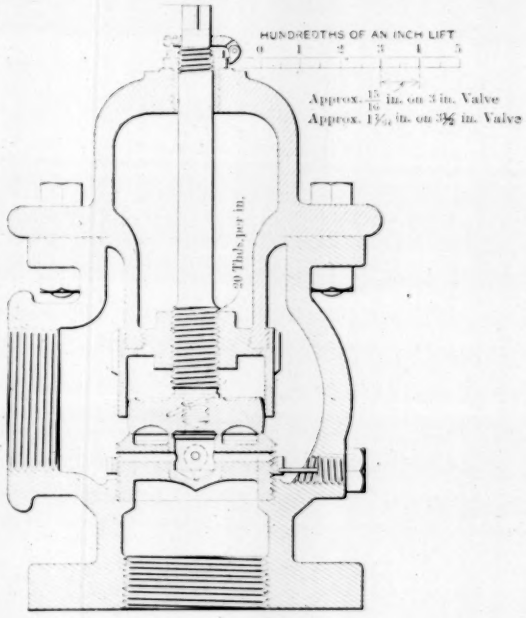


FIG. 2 CROSBY FLAT SEAT INSPECTOR POP SAFETY VALVE

gage attached. The boiler gage used was a Crosby standard test gage attached to the 10-in. pipe. In nearly all of the tests this gage was read at 1-minute intervals. In recording the pressures averages were taken from five readings in each case.

6 The pressure coming on the valve tends to make the opening through the valve greater than the "lift as set," due to the yielding of the metal. To determine this yielding, the valves were placed in a small Riehlé testing machine and a load equal to the steam pressure on the bottom of the valve applied to the valve The additional opening due to this yielding was determined by micrometer measurements.

7 All tests with low lifts were made at least 60 minutes long to minimize any error due to a difference of level in the hot-well under the condenser.

8 The condensed steam is pumped out of the hot-well by a pump operated through a float and valve. A difference of about 1 in. is required ordinarily to make this float operate. One inch difference in level in the hot-well means an error of 9 lb. of water.

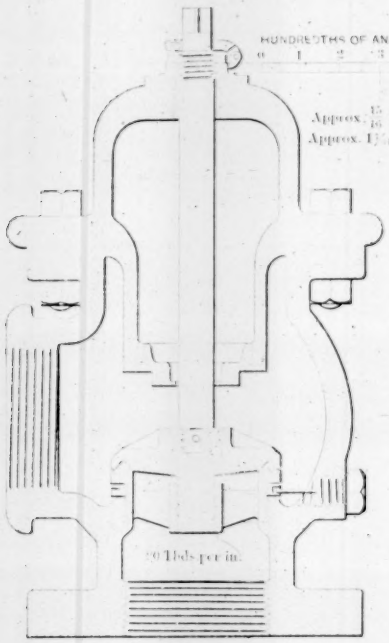


FIG. 3 CROSBY BEVEL SEAT INSPECTOR POP SAFETY VALVE

9 The different results are given both in tabular form (Tables 1 and 2) and in the shape of plots (Figs. 6 and 7). The different tests on one valve have been reduced to a common pressure by assuming the discharge through a given orifice to be proportional to the absolute boiler pressures. For such small variations in pressure as there were in these tests this assumption can introduce no error. Tables 1 and 2 give summaries only of results obtained in the tests. Tables were prepared, however, giving the results in detail, an example of which is shown in Table 3 which gives a part of the detailed records[1] summarized in Tables 1 and 2.

[1] Other of the records may be consulted in the rooms of the Society.

10 In the locomotive valves the muffler was screwed down to its lowest position. In the inspector valve flat seat the ring was screwed

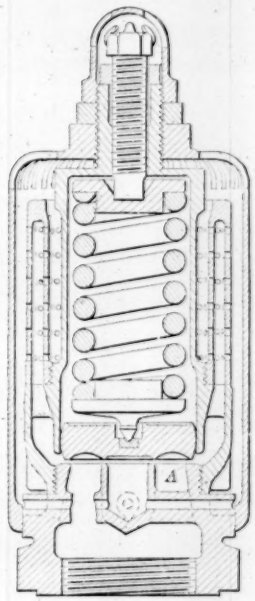


FIG. 4 SECTIONAL VIEW OF CROSBY MUFFLED LOCOMOTIVE POP SAFETY VALVE AS NOW MADE

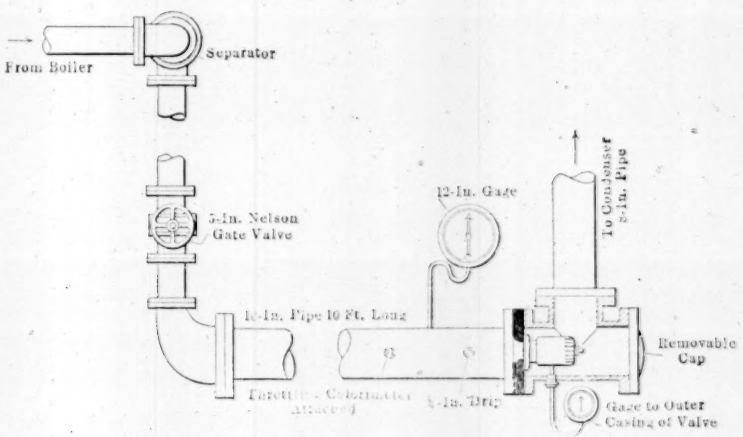


FIG. 5 ARRANGEMENT OF PIPING FOR SAFETY-VALVE TESTS

down so as to leave the holes half open. There was no ring on the bevel-seated valves.

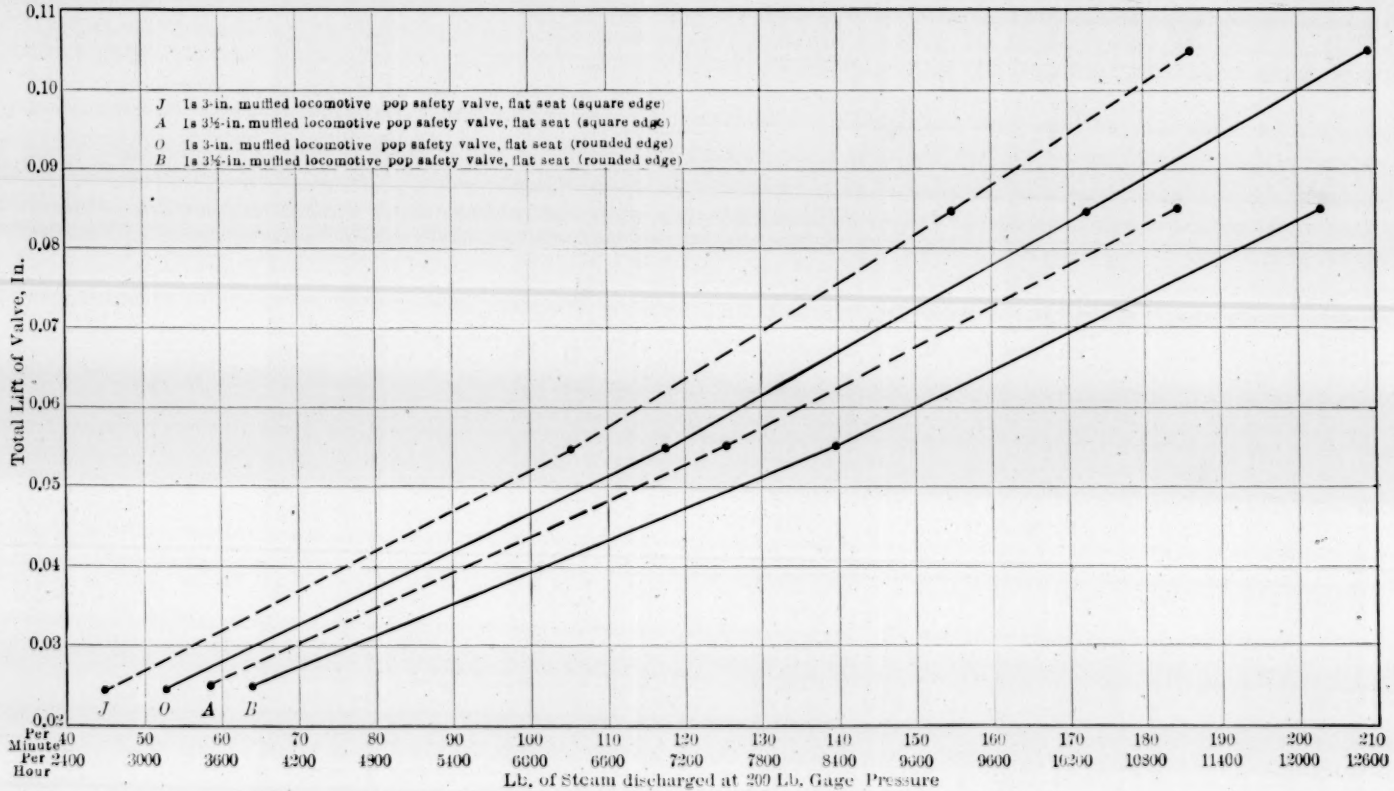

J 1s 3-in. muffled locomotive pop safety valve, flat seat (square edge)
A 1s 3½-in. muffled locomotive pop safety valve, flat seat (square edge)
O 1s 3-in. muffled locomotive pop safety valve, flat seat (rounded edge)
B 1s 3½-in. muffled locomotive pop safety valve, flat seat (rounded edge)
Total Lift of Valve, In.
0.11
0.10
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
J
O
A
B
Per Minute 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210
Per Hour 2400 3000 3600 4200 4800 5400 6000 6600 7200 7800 8400 9000 9600 10200 10800 11400 12000 12600
Lb. of Steam discharged at 209 Lb. Gage Pressure

FIG. 6 RESULTS OF TESTS ON CROSBY LOCOMOTIVE POP SAFETY VALVES

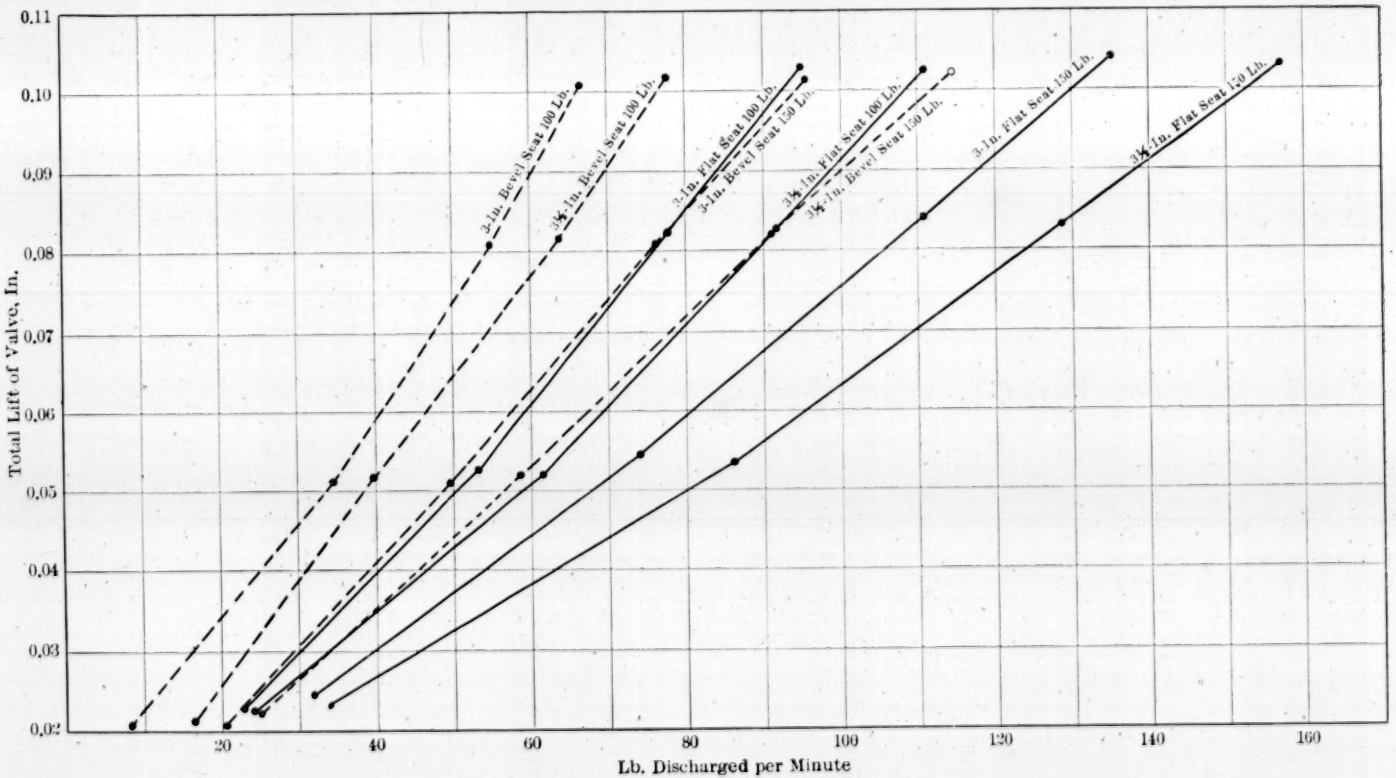


FIG. 7 RESULTS OF TESTS ON CROSBY INSPECTOR POP SAFETY VALVES

ACCURACY OF THE VARIOUS READINGS

11 The lift of the valves could be set with an accuracy as great as that of the 20-thread screw. As the total motion was only $\frac{1}{10}$ in.

TABLE 1 SUMMARY OF RESULTS ON CROSBY MUFFLED LOCOMOTIVE POP SAFETY VALVES

3-In. Flat Seat, Square Edge, Valve marked "J"

Lift of valve as set	0.02	0.05	0.08	0.10
Total lift including yield of metal	0.0247	0.0547	0.0847	0.1047
Lb. discharged per hr.	2845	6313	9342	11222
Boiler pressure gage	209.0	201.0	201.5	201.0
Lb. discharged per hr. reduced to 200 lb. gage	2743	6284	9277	11170
Lb. discharged per min. reduced to 200 lb. gage	45.7	104.7	154.6	186.2

3-In. Flat Seat, Rounded Edge, Valve marked "O"

Lift of valve as set	0.02	0.05	0.08	0.10
Total lift including yield of metal	0.0247	0.0547	0.0847	0.1047
Lb. discharged per hr.	3309	7250	10414	12442
Boiler pressure gage	209.0	205.8	201.3	198.3
Lb. discharged per hr. reduced to 200 lb. gage	3178	7059	10352	12542
Lb. discharged per min. reduced to 200 lb. gage	52.9	117.7	172.5	209.0

3½-In. Flat Seat, Square Edge, Valve marked "A"

Lift of valve as set	0.02	0.05	0.08	0.10
Total lift including yield of metal	0.0251	0.0551	0.0851	0.1051
Lb. discharged per hr.	3688	7688	11319	
Boiler pressure gage	212.3	204.2	205.0	
Lb. discharged per hr. reduced to 200 lb. gage	3485	7541	11062	
Lb. discharged per min. reduced to 200 lb. gage	58.1	125.7	184.4	

3½-In. Flat Seat, Rounded Edge, Valve marked "B"

Lift of valve as set	0.02	0.05	0.08
Total lift including yield of metal	0.0251	0.0551	0.0851
Lb. discharged per hr.	3989	8482	12172
Boiler pressure gage	209.2	202.7	199.5
Lb. discharged per hr. reduced to 200 lb. gage	3825	8377	12201
Lb. discharged per min. reduced to 200 lb. gage	63.7	139.6	203.4

it is probable that the error from this source is not over half a thousandth of an inch. The difference in expansion due to temperature between the body of the valve and the valve and its spindle was obviated in the case of the locomotive valves by making these

TABLE 2 SUMMARY OF RESULTS ON CROSBY INSPECTOR POP SAFETY VALVE

3-In. Flat Seat

Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0224	0.0524	0.0824	0.1024
Steam discharged per hr., lb.	1504	3473	4952	5998
Boiler pressure gage	111.8	109.8	106.3	105.9
Steam discharged per hr. reduced to 100 lb. gage.	1364	3200	4694	5703
Steam discharged per min. reduced to 100 lb. gage.	22.7	53.3	78.2	95.1
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0244	0.0544	0.0844	0.1044
Steam discharged per hr., lb.	1984	4627	6854	8438
Boiler pressure gage	159.6	157.9	155.9	156.1
Steam discharged per hr. reduced to 150 lb. gage.	1875	4415	6617	8137
Steam discharged per min. reduced to 150 lb. gage.	31.2	73.6	110.3	135.6

3-In. Regular Bevel Seat

Lift of valve as set	0.02	0.05	0.08	0.10
Lift of valve including yielding of metal	0.0207	0.0507	0.0807	0.1007
Steam discharged per hr., lb.	879	2219	3500	4412
Boiler pressure gage	110.4	109.2	108.3	109.8
Steam discharged per hr. reduced to 100 lb. gage.	806	2054	3264	4065
Steam discharged per min. reduced to 100 lb. gage.	13.4	34.2	54.4	67.8
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0210	0.0510	0.0810	0.1010
Steam discharged per hr., lb.	1313	3049	4670	5788
Boiler pressure gage	161.9	154.9	152.2	151.3
Steam discharged per hr. reduced to 150 lb. gage.	1225	2961	4608	5743
Steam discharged per min. reduced to 150 lb. gage.	20.4	49.4	76.8	95.7

3½-In. Regular Flat Seat

Lift of valve as set	0.02	0.05	0.08	0.10
Lift of valve including yielding of metal	0.0222	0.0522	0.0822	0.1022
Steam discharged per hr., lb.	1479	3636	5408	6760
Boiler pressure gage	105.2	99.2	98.1	102.1
Steam discharged per hr. reduced to 100 lb. gage.	1415	3661	5497	6638
Steam discharged per min. reduced to 100 lb. gage.	23.6	61.3	91.6	110.6
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0232	0.0532	0.0832	0.1032
Steam discharged per hr., lb.	2070	5236	7772	9286
Boiler pressure gage	153.3	152.8	150.8	147.1
Steam discharged per hr. reduced to 150 lb. gage.	2029	5148	7734	9452
Steam discharged per min. reduced to 150 lb. gage.	33.8	85.8	128.9	157.5

TABLE 2—Continued

3½-IN. BEVEL SEAT

Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0215	0.0515	0.0815	0.1015
Steam discharged per hr., lb.	1058	2488	3878	4788
Boiler pressure gage	110.2	105.0	102.5	102.4
Steam discharged per hr. reduced to 100 lb. gage	972	2384	3795	4690
Steam discharged per min. reduced to 100 lb. gage	16.2	39.7	63.2	78.2
Lift of valve as set	0.02	0.05	0.08	0.10
Lift including yielding of metal	0.0222	0.0522	0.0822	0.1022
Steam discharged per hr., lb.	1504	3518	5616	6746
Boiler pressure gage	152.2	151.1	154.9	147.4
Steam discharged per hr. reduced to 150 lb. gage	1484	3495	5454	6854
Steam discharged per min. reduced to 150 lb. gage	24.7	58.2	90.9	114.2

metals the same. In the case of the flat-seated inspector valves the body and spindle were of cast iron; in the case of the bevel-seated inspector valves the spindle was of composition, but the valve lift was adjusted by the disc-guide threaded below the seat. It will be seen by referring to the drawings that these combinations caused this difference of expansion to be reduced to a minimum.

TABLE 3 3-IN. MUFFLED LOCOMOTIVE POP SAFETY VALVE

FLAT SEAT WITH ROUNDED EDGE, VALVE MARKED O, 0.08 IN. LIFT AS SET, JUNE 16, 1910

Time	Tank, Weight	Weight of Condensed Steam	Average Boiler Pressure (Gage)	Pressure Muffler (Gage)
10:24	232			
	1120			
10:29		888	204	0
	275			
	1133			
10:34		858	198	0
	261			
	1131			
10:39		870	204	0
	266			
	1155			
10:44		889	206	0
	264			
	1105			
10:49		841	197	0
	265			
10:54	1126	861	199	0

Total time, 30 min.; total steam, 5207 lb.; steam per hour, 10,414 lb.; average boiler gage 201.3 lb.; quality of steam (dry steam =1) =0.996.

12 In testing the valves the adjustments and settings were always made while the valve was hot, practically at the temperature of the valve during a test.

13 The additional lift due to the yielding of the metal because of the presence of steam pressure on the under side of the valve was determined in the case of each valve. The valve was supported by its flange in a testing machine and a load corresponding to 100, 150, or 200 lb. per sq. in. applied to the valve. The movement of the valve with reference to its seat was measured by a micrometer caliper. Different sets of readings on any one valve varied 0.0005 in. and the results are liable to be in error by this amount. It is probable that considering all of the errors in determining the total lift of the valve the result is good to about 0.001 in.

14 The steam was condensed in an Alberger bottom inflow surface condenser with a hot-well at the bottom. This condenser is used in connection with a 500-kw. Parsons turbine in the engineering laboratories at the Massachusetts Institute of Technology. Besides the regular drain to the hot-well there were additional drains of 4-in. pipe, one from each end of the condenser to the hot-well. The level in the hot-well varied about 1 in., which corresponds to about 9 lb. of steam.

15 Each weighing of a tank, empty or full, is good to $\frac{1}{2}$ lb. In the runs with 0.08 and 0.10 in. lifts, if all errors in weighing are assumed to be cumulative and an error of 1 in. in the level in the hot-well be also considered, the maximum error is 15 lb. The probable error is less but even this is a small percentage of the total.

16 In the case of the 0.02 in. lift, the maximum possible error due to all sources would be 10 lb. and for the very worst case this is $\frac{10}{586}$ or less than 2 per cent. The results of the calorimeter tests are good within 2 in the third decimal place.

17 The pressure readings by the boiler gage were taken at 1-minute intervals during most of the time, but occasionally more frequently when the pressure was varying. The gage was tested before and after each series of tests and was found correct.

ADDENDUM BY A. B. CARHART [1]

18 This series of tests of safety valves is notable in that the measurements of discharge are quantitative instead of merely com-

[1] A. B. Carhart, Supt. Crosby Steam Gage & Valve Co., prepared additional data upon the type of valves tested and an abstract from this manuscript is herewith published.

parative and that the steam is measured by the condensation of the discharge, with determination by calorimeter of the moisture of the steam entering the valves. Some idea of the large condenser and boiler equipment required may be gained when the total discharge figures are compared with average conditions in ordinary work. And this tremendous rate of steam discharge was not for a few minutes only, but was maintained through 1-hour and ½-hour periods of uniform flow.

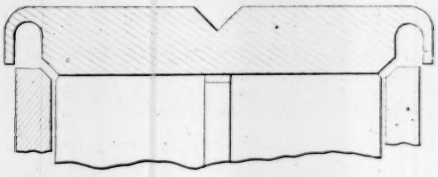


FIG. 8 BEVEL-SEATED VALVE

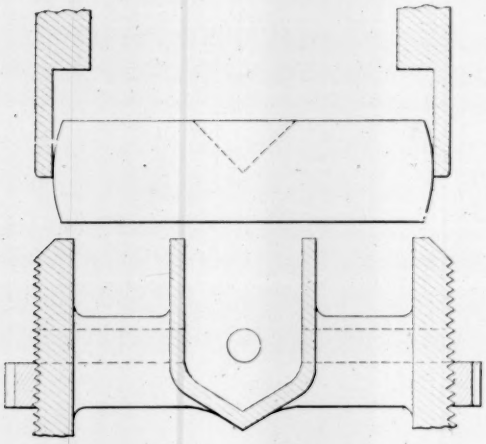


FIG. 9 ANNULAR VALVE

19 Every precaution was observed to avoid errors and all the readings were made by Professor Miller personally. The work was carried on at the Massachusetts Institute of Technology, because no commercial plant was found where the large steam supply could be uniformly maintained under absolute control. A part of the laboratory equipment there was utilized, but it was found necessary to erect a considerable amount of special apparatus and piping, and this preparation for the tests extended over nearly a year.

20 As the valves were adapted from ordinary commercial articles, altered in a few details to meet the requirements of convenient orifice regulation as planned by Professor Miller, it may be of importance to note what these peculiarities of construction were, and to supply some of the essential measurements that will be of interest to those who would like to make their own calculations of the areas of the valve passages and orifices.

21 Two of the iron-body valves were of the bevel-seated type, the contacting seat and disc faces forming a single steam-tight joint at an angle of 45 deg. to the vertical. The other two iron-body valves were of the annular type with two concentric seats in the horizontal plane, hence called flat-seated annular valves. The four locomotive valves were all of the flat-seated annular type, but in two of them the inner edge of the larger seat face was made slightly rounded, differing in this detail from the other two similar locomo-

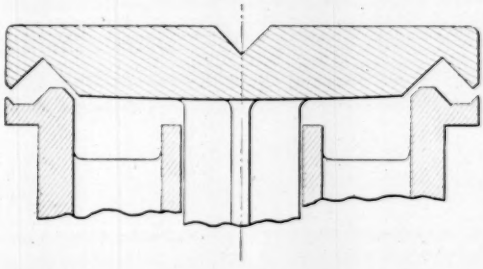


FIG. 10 TYPE OF BEVEL-SEATED VALVE USED IN TESTS

tive valves with which they were to be compared. The effect of this slight change of orifice form and proportion in the locomotive valve seats in greatly increasing the steam discharge is clearly demonstrated.

22 Pop safety valves were invented about 60 years ago, and about 30 years later were perfected in this country and went into general use. Broadly, the invention consists of an addition to the disc of the safety valve, so designed that when the valve is closed such addition is excluded from the action of the steam, but when the valve opens the outflowing steam acts upon it and with the initial force causes the valve disc suddenly to rise higher and the spring to be more compressed than it would be with a force due to the steam pressure upon the original area only; thus there is afforded a prompt,

full-open passage at the seat through which the maximum quantity of steam will flow at once and give relief to the boiler.

23 The bevel-seated valves (Fig. 8) generally have this additional area at the periphery of the disc, outside of the seat, where it forms a chamber with more or less contracted outlet at the extreme lip of the valve, through which all the steam must pass after escaping over the seat before it reaches the open air. As the seat is formed at an angle of 45 deg. to the vertical, the passage between the seat and disc faces when the valve opens is diagonally upward, and what is called the lip of the valve is so related to this seat that the steam is forced to impinge directly upon it before being deflected downward toward the bottom of the chamber where it reacts, the effect of the impact upon the lip of the valve being increased by the expansive force of the steam acting in the lip chamber on account of the partial and momentary confinement before it passes into the open air. This force is regulated in various ways. Sometimes openings are made through the lip or in the floor of the chamber, through which the steam escapes and so reduces the pressure within the lip chamber that the valve disc will not lift too high. Most valves of this kind have a threaded ring surrounding the chamber, providing a micrometer adjustment by means of which the opening at the lip may be made wider or narrower at will. The ring sometimes slightly overlaps the disc at its periphery until the valve has lifted an appreciable amount.

24 Annular valves (Fig. 9) have the additional area located at the center of the disc and within the outer seat. This area is excluded from the action of the steam when the valve is closed, by means of an inner seat, but acted upon directly by the steam when the valve opens. The disc is simply a flat cover for the valve base and has two concentric seats in the same horizontal plane. The central area of the disc, which is not exposed to the steam when the valve is closed, covers a well which communicates to the outer air through four hollow arms radiating from it out through the sides of the valve body. When the valve opens, the steam is discharged directly across the outer flat seat, and there is also a separate additional flow over the inner seat into the central well and out through the four hollow arms. The openings through these arms are controlled by a sleeve, threaded on the valve body, so that it may be turned upward or downward as desired. If it should close the open ends of the hollow arms, preventing the escape of the steam through

the central well, the reactionary effect upon the disc to lift it against the spring would be greater than if the openings from the well were unobstructed; and by locating the sleeve at intermediate points the lift of the valve can be varied.

25 While both styles of valves are efficient, it will be noted that in the bevel-seated type all of the steam passes over the single seat and through the additional regulating chamber at the lip of the valve, and in some cases through a constricted passage at the periphery of the lip; while in the annular valve the steam for the relief of the boiler pressure passes directly through the outer seats without any obstruction whatever, unaffected by any regulation of the by-passed steam through the center that may be necessary. Also, for a given lift of valve, the area of opening through a flat-seated valve is nearly $1\frac{1}{2}$ times that through a valve with bevel seats.

26 In bevel-seated valves the disc is ordinarily guided vertically so that it will return properly to its seat when the valve closes, by means of four or more radiating wings or vanes depending from the face of the disc in the steam space and bearing against the sides of the inlet or throat of the valve (Fig. 8). In the bevel-seated valves tested by Professor Miller, the disc was mushroom-shaped, the guide being small in diameter and acting in an open hub supported in the steam space by arms extending from the sides (Fig. 10). But in the annular valve there are no guides or wings attached to the disc, which is simply a flat member having the truncated form of the central zone of a sphere. This is placed in a cylindrical chamber outside and above the steam space, and there it lifts or moves freely as acted upon by the steam against its lower face and by the opposed spring which presses upon it at its center. The contact between the chamber and the spherical part of the disc is the least possible, which permits universal movement of the disc without possibility of cocking or sticking, and yet insures that the disc will return accurately upon its seats when the valve closes.

27 In the two of the iron-body valves that were fitted with these annular seats, and in two of the locomotive valves, the escaping steam was compelled to make a sharp right-angled turn in passing out over the inner edge of the main discharge seat (Fig. 11). This had been the construction for many years, as the concession to more simple manufacturing operations. It was proposed by Professor Miller to round this corner slightly, as indicated at A' in Fig. 12, at the junction of the side wall of the inlet or throat of the valve with

the main outer seat, for he believed that this would permit a larger flow of steam. Therefore, two other locomotive valves were submitted to test, exactly like the first in all respects except that this edge was rounded, but only to the small extent that the castings would permit without change in patterns. The valves with the angular corner are referred to in the tests as having the "square edge," while the valves with the rounded corner are designated as having the "rounded edge." Fig 4 shows the later form of these valves with shape of inlet passage modified.

28 The locomotive valves were cast with flanged bases, similar

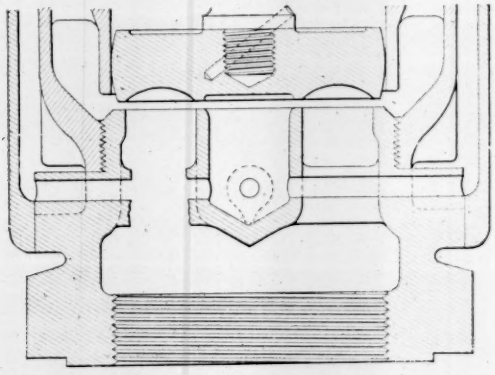


FIG. 11 SECTIONAL VIEW OF ANNULAR VALVE

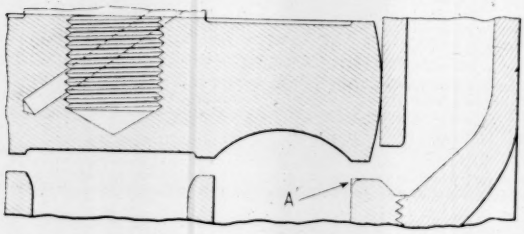


FIG. 12 ANNULAR VALVE SHOWING ROUNDED CORNER OF VALVE SEAT

in size to the iron-body valves, to suit the pipe connections to which they were attached during the tests. The designating letters, *J*, *O*, *A*, and *B*, were stamped upon the locomotive valves before the tests, simply as a convenient means of identification.

29 In both bevel-seated valves, the total width of the seat-face *CBE* (Figs. 13 and 14) was 0.088 in., and the vertical distance *m* between the planes of the lower edge of the seat face at *C* and

the outer edge of the seat face at E measured 0.062 in., in both the 3-in. and $3\frac{1}{2}$-in. valves; therefore when the valve disc was lifted 0.02 in. and 0.05 in., in the tests, the condition is represented by Fig. 13, and when the disc was lifted 0.08 and 0.10 in., the lower face of the disc at A was above the plane of the outer edge of the seat face at E and the condition is represented by Fig. 14. It will be seen of course that the entrance to the diagonal passage through the seat when the valve is open is measured on the perpendicular line let fall from A to the seat face at B, and this has the mean diameter d, which is greater by the distance $\frac{\text{lift}}{2}$ than the diameter D, which measures the area of the disc that is under pressure when the valve is closed.

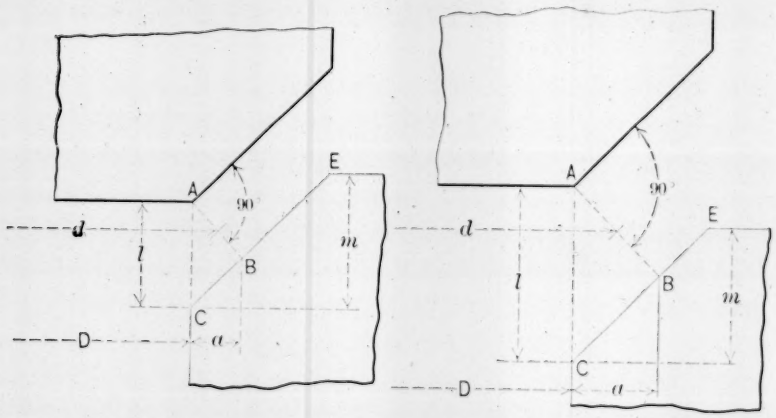
30 It is apparent that calculating the area of the discharge passage upon the greater diameter d instead of the distance D gives some theoretical advantage in this style of valve, yet the total quantity of steam discharged is nevertheless not as great as in the case of the flat-seated valves, in which the free, open passage through the outer seat is measured by the normal circumference of the inner edge of the seat having the diameter D, multiplied by the actual vertical lift of the disc.

31 In the annular valves, the area of the opening into the central well over the inner seat is considerably greater than the outlet area from the well through the four hollow regulating arms, even when these passages are left full-open; for the smaller edge of the inner seat face leading to the central well is 1 1/16 in. in diameter in the 3-in. valve, giving a circumference of 3.33 in., and in the $3\frac{1}{2}$-in. valve the smaller edge of this inner seat over which the steam passes to the central well is $1\frac{1}{4}$ in. in diameter, giving a circumference of 3.92 in.; so that, at 0.08 in. lift for example, the area of the entrance to the well is 0.26 sq. in. in the 3-in. valve and 0.31 sq. in. in the $3\frac{1}{2}$-in. valve. Compared with this, the hollow arms, leading from the central well, are $\frac{1}{4}$ in. in diameter in the 3-in. valve and 9/32 in. in the $3\frac{1}{2}$-in. valve, the four arms giving an additional possible total area of 0.196 sq. in. to be added to the main discharge in the 3-in. valve, and 0.248 sq. in. additional in the $3\frac{1}{2}$-in. valve.

32 Measurements of the pressure in the central well chamber, however, made by connecting a pressure gage to it by means of a separate pipe, show that the pressure existing there when the valve is blowing is appreciably less than the indicated boiler pressure. The

total effective discharge area to be credited to these auxiliary outlets through the central well may therefore be taken as from 4/10 to 8/10 of their nominal measured area just given.

33 In calculating the total effective discharge area of these annular valves, first multiply the smaller circumference of the outer seat by the full vertical lift of the disc and add to this a reduced percentage of the area of the four arms or of the portion of their exit ends not covered by the regulating gate. The effective pressure in these restricted passages is of course considerably reduced, and depends more or less upon the amount of the opening at the seat leading into this central chamber and the freedom of flow into it, for



FIGS. 13 AND 14 DIAGRAMS SHOWING DISCHARGE AREA OF BEVEL-SEATED VALVES

when the disc is lifted from its seat only 0.02 in., there must be an appreciable restriction or wire-drawing of the steam flowing into the central well, and therefore it seems fair to take the value of the discharge area of the hollow arms under these conditions as only 40 per cent of their measured area, increasing to 70 per cent when the lift is 0.05 in. and 80 per cent for the normal lifts of 0.08 and 0.10 in. But these approximations are not important and do not greatly affect the results, for the function of this small by-passed flow of steam is principally to regulate the lift of the disc, and the quantity of discharged steam through the central passages is only incidental and additional, and for all practical purposes these estimated values may be taken as fairly conservative.

34 The width of the seat face on the casting of the base of the

annular valve is approximately 3/16 in. for the outer seat and $\frac{1}{8}$ in. for the inner seat, and the faces on the disc fairly coincide with these when the valve is closed, though the actual width of the contacting faces may be better assumed as about $\frac{1}{8}$ in. for the outer seat and 3/32 in. for the inner seat leading to the central well. The formula for calculating the effective discharge area of the annular valve is:

$$\begin{aligned}\text{Discharge area} &= (\text{circumference of outer seat} \times \text{lift}) \\ &\quad + 4\times (\text{net effective area through each arm}) \\ &= \pi Dl + 0.8 \times \pi(\text{diameter of arm})^2\end{aligned}$$

35 In the bevel-seated valves tested by Professor Miller, the area of the discharge passage over the valve seat may be calculated by the well-known formula, which is made clear by reference to Figs. 13 and 14 illustrating the two cases involved, one where the disc at A does not lift higher than the vertical distance m between the inner and outer edges C and E of the seat face, and the other where the lower face of the disc at A rises above the horizontal plane of the outer edge E of the seat face.

$$AB = l \sin 45 \text{ deg.} = 0.707l$$

$$d = D + \left(2 \times \frac{a}{2}\right) = D + \frac{l}{2}$$

$$\text{Seat circumference} = \pi D$$

$$\text{Orifice circumference} = \pi\left(D + \frac{l}{2}\right)$$

$$\begin{aligned}\text{Discharge area} &= (AB)\pi\left(D + \frac{l}{2}\right) \\ &= (0.707l)\pi\left(D + \frac{l}{2}\right) \\ &= 0.707\pi Dl + \frac{0.707\pi l^2}{2} \\ &= 2.22Dl + 1.11l^2\end{aligned}$$

This form of the equation gives results about $1\frac{1}{4}$ per cent larger than the usual approximation

$$\text{Discharge area} = 0.707\pi Dl$$

36 In both types of valves concerned in these tests, the lower end of the inlet connection measures appreciably larger than the actual diameter at the valve seat, and the net area of the unobstructed passage leading up to the edge of the valve seat where the steam escapes is several times greater than the largest possible area through the seat for steam discharge that is intended to be opened in the ordinary operation of the valves. In the case of the annular valves the area

of the inlet passages is approximately three times that of the discharge passages and in the case of the bevel-seated valves it is from three to four times, assuming in each case that the valve lift is 0.1 in.

37 The accompanying chart, Fig. 15, shows the relative areas in square inches of the narrowest parts of the passages and chambers through which the steam discharged, in the locomotive valves, after passing the valve seat; and the progressive increase explains why the valves were satisfactorily muffled without any retarding effect upon the escaping steam, and why, as noted by Professor Miller, no back pressure could be detected.

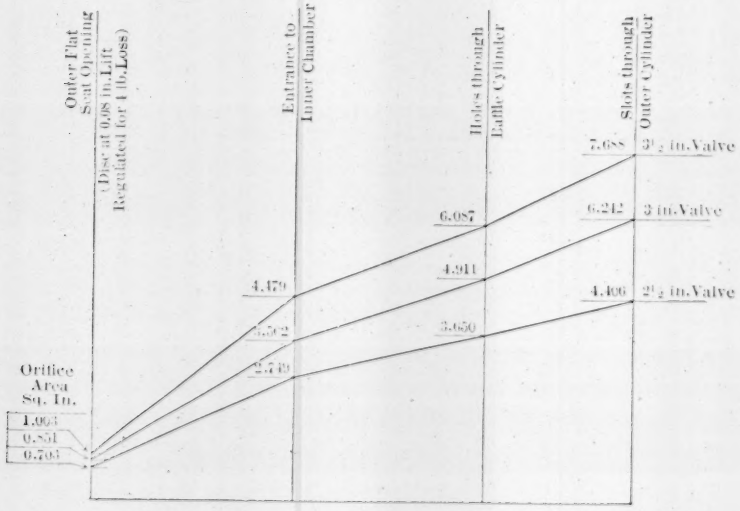


FIG. 15 DIAGRAMS SHOWING PASSAGE AREA AT DIFFERENT POINTS IN THE LOCOMOTIVE VALVES

38 When any safety valve is closed steam-tight, the actual line of contact between the disc face and seat face is a sinuous line without appreciable width, which at points approaches the inner or outer edge of the seat face as happens. Therefore the exposed area of the valve disc, under pressure when the valve is closed, is never exactly the same as the theoretical area calculated by the rated diameter of the valve seat. But for all practical purposes it is nearly so and the diameter of the inner edge of the seat is generally taken as the measure of the disc area.

39 This method of measurement shows a total of 7.07 sq. in.

area of the disc under pressure in the 3-in. bevel-seated valve, when the valve is closed, and 9.62 sq. in. in the $3\frac{1}{2}$-in. valve. In the flat-seated annular valve the net area of the annular face under pressure is 5.79 sq. in. for the 3-in. valve and 7.85 sq. in. for the $3\frac{1}{2}$-in. valve. To obtain these values the areas enclosed within the central seats were deducted. In the 3-in. valve the diameter of the central seat is 1 9/32 in. in diameter and in the $3\frac{1}{2}$-in. valve $1\frac{1}{2}$ in.

40 Multiplying any of these effective areas by the boiler pressure will give the load upon the disc at the moment before the valve opens, which load must be sustained by the valve spring. This is calculated to be of the proper proportions to hold the valve to its seat until the opening pressure is reached and after permitting a full opening of the valve by the lifting of the disc against the spring pressure must return the disc again to its seat and close the valve when the predetermined amount of decrease in boiler pressure has been attained. If the total area of the disc be taken into account, after the valve opens, it is easy to determine the mean pressure of the discharging flow effective to raise the disc, until the reaction of the spring is sufficient to close the valve against the reduced pressure upon the whole face of the valve disc.

41 Assuming, for example, a compression of the spring of 0.50 in. from its free length to exert a load upon the disc equal to the steam pressure of 100 lb. per sq. in., then a lift of the disc of 0.10 in. further would increase the tension of the spring and increase the reactionary load 1/5 more than when the valve is closed. Therefore the discharging steam acting by impact and expansion against the whole area of the disc face when the valve is blowing would have to sustain momentarily a maximum load 20 per cent greater than was exerted by the steam at 100 lb. per sq. in. upon the initial area of the disc when the valve was closed, although after the steam has passed out over the valve seat into the lip chamber or into the central well it is of course at much lower pressure and has increased considerably in volume.

42 The extreme diameter of the outer edge of the lip of the disc in the 3-in. bevel-seated valve may be taken as $4\frac{1}{8}$ in. when the valve is open, and $4\frac{3}{4}$ in. in the $3\frac{1}{2}$-in. valve. The face of the outer seat of the disc in the annular valve has an appreciable width, making the extreme diameter of the disc acted upon by the steam when the valve is open 3 3/16 in. in the 3-in. valves and $3\frac{3}{4}$ in. in the $3\frac{1}{2}$-in. valves.

43 Although the discharge passage over the outer flat seat of the annular valve is nearly $1\frac{1}{2}$ times as large as is possible in any bevel-seated valve of the same diameter-size and lift, the area of the valve disc exposed to the steam when the valve is closed is however only about four-fifths of the total area of the disc, by reason of the central portion within the inner seat circle being shut off from the boiler pressure until the valve opens. This causes less load upon the disc, much less strain and pounding of the valve in seating and less necessary size and spring trouble, than in a bevel-seated valve of the same rated size or seat diameter. When the valve is closed, as the steam pressure of the boiler acts only upon the annulus, or the area between the outer and inner seats, the spring is required only to meet this steam pressure. Therefore the spring has to exert less absolute force than if it held the disc to its seat against the steam acting upon the whole area of the disc. The universal action of the disc due to its spherical edge within the cylinder provides against any binding effect from undue eccentricity of the spring, for all springs in compression necessarily shorten with a torsional movement and side-thrust transmitted to the disc-guides.

44 There is a decided advantage, therefore, in reducing the necessary movement of the spring in its compression and distortion when the valve opens to the minimum amount possible, while yet attaining the required volume of steam discharge.

COMMERCIAL APPLICATION OF THE TURBINE TURBO-COMPRESSOR

By Richard H. Rice, Published in The Journal for March, 1911

ABSTRACT OF PAPER

The paper describes a compressor unit for blast-furnace work consisting of a four-stage Curtis steam turbine direct-connected to a six-stage centrifugal compressor. This was built and installed at Oxford Furnace, N. J., by the General Electric Company, and is the first of this type of apparatus to be made in this country. It is governed automatically by mechanism which depends for its action upon a change in the rate of air flow, by which means the speed of the turbine is varied as required. The governor regulates the volume of air delivered per minute so as to keep the rate of discharge constant at the value determined by the furnace superintendent. It is found that on account of the steadiness of operation and more uniform conditions the output of the furnace has been increased and it is concluded that, considering all the factors, the centrifugal compressor as a blowing apparatus can be operated at a lower net cost than any other means for blowing furnaces. The paper contains data upon sizes and capacity of the apparatus with diagrams showing the characteristics of its performance.

DISCUSSION

E. D. Dreyfus (written). The comparison of the ultimate economies of the reciprocating gas-engine blowing engine, and the high-pressure turbine-driven compressor is most vital. But we must not allow ourselves to be guided by any abstract quantities, and in order to obtain a comprehensive idea of their relative commercial value, we must study a wide range of sizes and conditions. The relation of single small sets of the two types will in all probability be reversed when installations of a great many large units are compared. Isolated data should invariably be treated guardedly, more particularly where an appreciable variation in conditions may exist. There is one thing which occurs to my mind especially and that is

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers. All discussion is subject to revision.

the large gas-engine blower will have but little showing in the extensions of any present mills where it will be practical to install and operate low-pressure turbine-driven compressors in connection with older steam-engine-driven tubs. The investment value of this earlier apparatus will thus be preserved and at the same time its economy when operating in connection with the low-pressure turbine will be greatly bettered. Another point of economic consideration is the coördinating of the low-pressure turbine with the gas plant, whereby the position of the gas engine may be materially advanced, an analysis of which was contributed by the writer to Power, April 11, 1911.

J. E. JOHNSON, JR. The efficacy of the control of the volume of air blown is the question which will arise in the mind of every practical furnace man as soon as he gives this subject consideration. Changes in quantity of air as small as 1 or 2 per cent are frequently made in the quantity of blast delivered to the furnace as one of the means of controlling its operation and this makes it obvious that exact control is a very necessary feature of any blast-furnace blowing apparatus.

I had the pleasure of seeing the installation of the turbo-blower at the Port Henry plant of the Northern Iron Company about a year ago, and the simplicity of the governor is certainly admirable, but is a feature not exactly easy of comprehension to a man without a thorough mechanical training, and a prejudice against these machines is likely to exist in the minds of many furnace men.

It is a well accepted fact that the efficiency of the turbine is far higher at low pressures than it is at high, while on the other hand it is equally well known that for high pressures only, the reciprocating engine has a considerable advantage over the turbine. The result is that the most economical form of steam-driven apparatus is a combined unit in which the steam expands down to about atmospheric pressure in a good reciprocating engine, and is then delivered to a turbine on its way to the condensers. The very remarkable results obtained with this combination by Mr. Stott at the Interborough plant in New York have given, in the opinion of many, a decided check to the development of the gas engine.

In a similar way it has been known for years that for very low pressures the centrifugal blower, of which the turbo-blower is simply a development, is more economical than a piston blower. On the other hand, for higher pressures the piston compressor is undoubtedly to be preferred, particularly in blast furnace work, on account of its operating as a meter as well as a compressor of the highest efficiency.

A condition of piston blowing engine design is that the air cylinder must be large enough for the greatest volume to be handled, and strong enough for the highest pressures attained, with the result of enormous size and very massive and expensive construction in modern blowing machinery.

In spite of their high cost it is notorious among operating men that really reliable blowing engines are exceedingly scarce. The problem of the valves is in itself one of great difficulty. It is almost impossible to get an air inlet valve with sufficient area to allow the cylinder to fill absolutely without heavy loss by suction, and at the same time one quick enough to give correct results at the speeds required of modern engines.

The turbo-blower, on the other hand, suffers from the disability that it must have stages enough to blow the highest pressure ever required, although this high pressure will not, in ordinary operation, be required one per cent of the time, the normal pressure being perhaps only half as great.

If, therefore, we make a turbo-blower to deliver air of a certain nearly constant, moderate pressure, and a blowing cylinder of comparatively small size, strong enough to deliver air to the desired maximum pressure, we shall have bettered the conditions of operation of both types of apparatus, increasing their efficiency and reducing their cost. In other words, the turbo-blower and reciprocating-compressor combined are more efficient than either apparatus alone, exactly as the engine-turbine combination is more economical in the consumption of steam.

For the best and cheapest blowing engine, therefore, we should have a combination of these two units, a turbine-driven turbo-blower, delivering air partly compressed to the compressing end of a steam-driven blowing engine, the steam from the cylinder of this engine driving the turbine before passing to the condenser.

It may be objected that this is a complicated unit, but this is not the case. Most large modern furnaces require three large reciprocating blowing engines of the best type to operate them. These are, in good modern usage, disconnected compounds—a high-pressure steam cylinder on one engine and a low-pressure on the next, etc.

A turbo-blower supplying air to two of these, compressed to 8 lb., would enable them to deliver as much air as three engines drawing air from the atmosphere, and the efficiencies which are guaranteed in the use of exhaust steam by the builders, combined with the guaranteed efficiency of the turbo-compressor, indicate that this can be done at

a decided saving in first cost and with an efficiency certainly no lower and probably higher than that of either unit alone. This does not make any allowance for the possibilities of spray intercooling between the turbo-compressor and the reciprocating compressors, which, in ordinary practice, would be good for about 4 per cent additional saving at a very low cost.

The advantage of maintenance as compared with reciprocating engines, is undoubtedly with the turbo. At the same time the magnitude of the alternations of stress in the reciprocating engines is greatly reduced by delivering to them partly compressed air, and the problem presented by their inlet valve gear, which has given so many bad hours to both furnace men and engine designers, is practically eliminated by delivering to them denser air under pressure.

The governing of such a combined unit is extremely simple. The ordinary governor on the engine unit does all that is necessary and no governor other than one to prevent racing is necessary on the turbine. The steam supply of the engine passes directly to the turbine. As long as all conditions remain constant the speed of the turbine will be unchanged, but if the pressure required by the furnace increases a little, more steam is admitted to the engine by its governor, and this increase in quantity speeds up the turbine to a slight extent and so causes it to deliver air compressed to a slightly higher pressure.

This increase automatically compensates for any small lag due to the greater load on the engine and for the lower volumetric efficiency of the piston blowing engine at higher pressure as compared with low.

If, on the other hand, the speed of the piston engine is deliberately increased to deliver more wind, the greater supply of steam to the turbine will enable it to deliver a proportionately greater quantity of air at virtually the same pressure. Of course, some hand control will be applied to the turbine to enable the receiver pressures to be adjusted to the best conditions, but once this is done other changes will seldom be required.

For these reasons I believe that the steam-driven blowing plant of the future should consist of a turbine-turbo, piston-engine, piston-compressor combination.

This combination will have its greatest field in applications to existing plants where, as is so often the case, the existing blowing engine, while old, is by no means worthy of the scrap-heap, but is unable to deliver quite the pressure or quite the volume demanded by modern conditions. In such a case the introduction of a turbine-

turbo, taking steam from the engine and supplying pre-compressed air to it, will use what is now waste energy in the steam, and apply it in such a way as to increase both the capacity and the permissible pressure of the existing blowing cylinder.

A further advantage which the turbo has as the primary machine is that of an inlet free from pulsations. These are much more serious in inlet than in outlet, for the reason that the volume and velocity of the air are so much greater.

One of the serious problems which had to be overcome in the application of the dry blast was the construction of cooling chambers whose walls would not be thrown down by the pulsations produced by the piston compressor on the incoming air. The smooth inlet of the turbo will eliminate entirely difficulties arising from this source.

The necessity of massiveness both in stationary and in reciprocating parts, and the conditions of operating the air valves, have limited the best sizes of blowing engine so that these are not increasing in size. In fact, those now being built are commonly no larger than some constructed ten or twelve years ago.

The turbo-blower, on the other hand, lends itself admirably to enormous capacities in the same way that the turbine is commonly built for certain classes of service in sizes that have never been approached in reciprocating engines. For this reason it will not only be possible but advisable to install at large plants one turbo to supply several reciprocating engines, all of which, of course, will exhaust to it. This will contribute still more to low first cost, small space requirements and small maintenance charges.

I wish to express in conclusion my conviction that the introduction of the turbo-compressor marks the beginning of a better era for the mechanical equipment of blast furnaces, but that its best and easiest application will be in conjunction with rather than in supplanting piston blowing engines.

R. N. EHRHART.[1] The Westinghouse Machine Company have for some years been experimenting with centrifugal air compressors at their East Pittsburgh works, and as a result have developed a type of blower possessing features particularly adapting it to certain classes of work.

As the single-stage blower is the basic element of the multi-stage blower, analyses of the primary considerations should be applied to

[1] The Westinghouse Machine Company, East Pittsburgh, Pa.

the single part or stage which is duplicated in certain instances to make the multi-stage outfit. Any features pertaining to the generic element pertain to the whole, so that studies of the single-stage blower are more enlightening from a practical and educational standpoint. The following facts were brought out in the development of the elements of centrifugal compressors:

Centrifugal compressors having diffusion vanes in the part of the casing receiving the air from the rotating element have as an in-

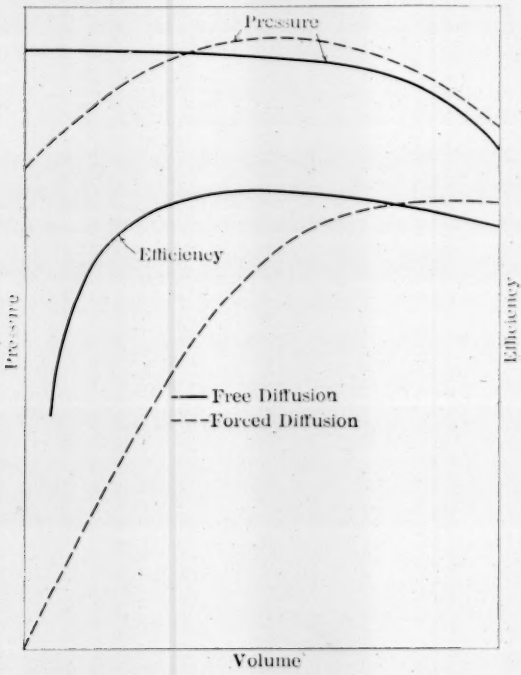


FIG. 7 COMPARISON OF BLOWERS WITH FREE AND FORCED DIFFUSION

herent characteristic the pressure curve shown in broken line in Fig. 7. (In this discussion, the writer calls this the forced-diffusion type of blower.) The efficiency also has the characteristic as shown in Fig. 7; that is, a maximum efficiency of approximately 74 per cent at rated volume and approximately 60 per cent at one-half volumetric rating. These results were obtained by extensive experimenting at East Pittsburgh and are approximately verified by Fig. 2.

For the industrial application of constant-speed blowers, it was early realized by the Westinghouse Machine Company that the

pressure characteristic as shown by the broken line (forced diffusion) in Fig. 7 was unsuitable, as reference to Fig. 8 clearly shows.

The forced-diffusion type of blower has a pressure characteristic with a well defined hump; that is, the pressure rises as the volume delivered increases to a certain point, and the pressure then rapidly falls off. In other words, for a given pressure there are two definite volumes that may be delivered. Referring to the lower set of curves, Fig. 8, at the pressure represented by the horizontal line there may be two volumetric deliveries, *B* and *D*. It is perfectly evident that blowers of the forced-diffusion type will be unsatisfactory for parallel operation, unless equipped with a complicated governing mechanism, since momentary disturbances may cause fluctuating line pressures

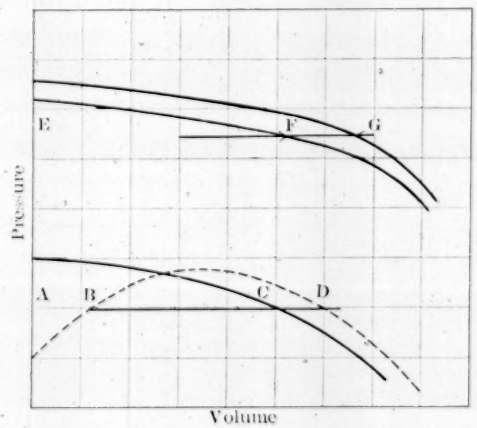


FIG. 8 CHARACTERISTICS REQUIRED FOR PARALLEL OPERATION OF BLOWERS

and if the disturbance has a magnitude measured from the horizontal line to the peak of the pressure characteristic, the volumetric delivery may vary from *B* to *D*. This latter possibility will in turn accentuate any line disturbances, causing still greater variation or fluctuation in volumetric delivery, assuredly at the sacrifice of efficiency.

In the case of parallel operation of alternators, we require a gradual drop in speed from no load to full load for stability. In the parallel operation of blowers, we should have the analogy of a gradual drop in pressure from no delivery to full delivery.

Blowers having this characteristic will work perfectly in parallel. Referring to Fig. 8, the upper pair of lines, shows the sharing of the load. One blower at the pressure represented by the horizontal line

will have a volumetric delivery *EF*, the other one a volumetric delivery *EG*, and fluctuations of pressure over a wide range will still insure relatively equal division of the load.

Positive blowers have the drooping characteristic at constant volume, and it is seen that if this is superimposed on the forced-

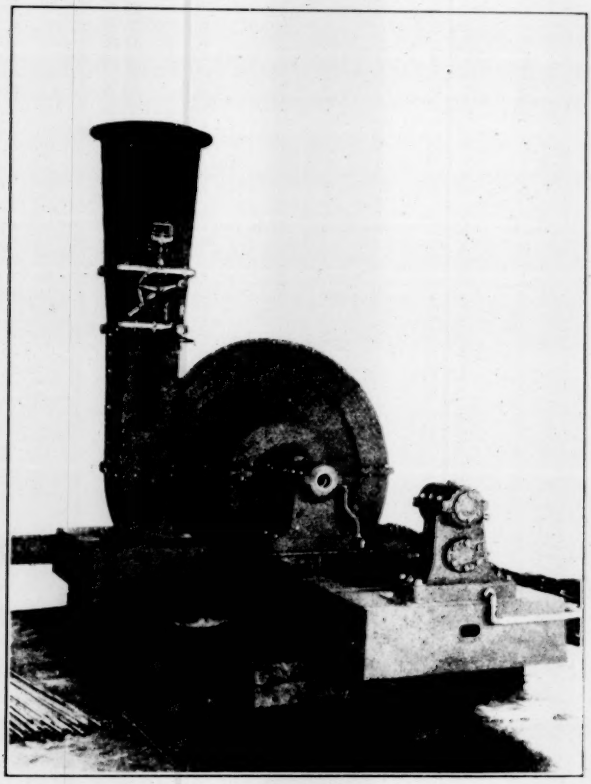


FIG. 9 A 600-H.P. BLOWER OF FREE-DIFFUSION TYPE

diffusion pressure characteristic, successful parallel operation cannot be had.

Of course, by resort to complicated speed-control mechanisms on what might be a constant-speed outfit, the forced-diffusion type of blower can be made to give the drooping characteristic.

Thorough experimenting has shown that by suitably constructing the stationary and rotating parts of the blower, a drooping pressure

characteristic can be obtained; that is, the pressure at no delivery is a maximum, and it gradually drops off to normal at normal rating. This characteristic is shown in full line in Fig. 7. It has also been found that the efficiency at fractional deliveries was wonderfully improved. As a matter of fact, the diffusion vanes in the forced-diffusion type of blower are a positive detriment at fractional loads, since their angle and shape can be made only to suit one discharge rate.

For example, in a blower with radial vanes, the air leaves the rotor at an angle of about 14 deg. with the tangent at normal delivery. At 50 per cent of normal delivery the angle is about 7 deg.; at 10 per cent of normal delivery it is about $1\frac{1}{2}$ deg. It is apparent that if the receiving ends of the diffusion vanes are set for normal delivery, they are $12\frac{1}{2}$ deg. from the correct position for 10 per cent of normal delivery, or 7 deg. from correct position for 50 per cent normal delivery.

Experiments at East Pittsburgh show that free spiral diffusion can be obtained with substantially the same conversion efficiency at 25 per cent normal delivery as at full normal delivery. The discharge areas and contour of vanes and receiving sides of the spiral whirl chamber must coincide exactly with the dimensions as fixed by experiments and calculations. Relatively small variations from proper shapes cause loss of velocity conversion efficiency.

Fig. 7 shows comparative tests of the two types of blowers made at the Westinghouse works, the splendid pressure characteristic and superior efficiency of the free-diffusion type at fractional loads being clearly illustrated.

The free-diffusion type of blower does not have the pulsation characteristic that the forced-diffusion type has. This is due to the total absence of diffusion vanes, so that free spiralic diffusion can adapt itself to any volumetric delivery. The pulsations as described by Mr. Rice in Pars. 22 to 25 (Fig. 4), would seem to be of such magnitude (approximately 9 to 10.5 lb.) as to make parallel operations difficult when they are in evidence. The method used at Oxford to get over the pulsating effect, that is, throttling, may be successful when applied to high-pressure blowers, but when applied to blowers of low pressure the loss of a few inches of water pressure by throttling could not be considered as it would represent too great a proportionate loss in efficiency.

A 600-h.p. blower of the free-diffusion type has just been put into operation at Great Falls, Mont. The pressure characteristic corresponding to that given in Fig. 7 is attained with a constant speed,

and no governing devices to change the motor speed are necessary. The outfit is driven by a 750-h.p. synchronous, 60-cycle motor. Fig. 9 shows the blower by itself.

C. G. DE LAVAL. The development of the turbo-compressor in this country is of recent origin and without question the combination of the steam turbine and turbo-compressor produces an ideal unit. It appears, however, that its field will rather be the compressing of large than of small volumes. Not long ago it was considered unsuitable for anything except exceedingly low pressure, such as ordinary blowing work. Today we find turbo-compressors in use abroad for 150 lb. equal in thermal and mechanical efficiency to reciprocating compressors.

It is well to consider the distinction between a blower and a compressor, and it appears to the writer that the machine described is not a compressor, but a turbo-blower. Both may be of the same design, but this is not essential. The machine described is a multi-stage blower or fan, and apparently has no rubbing surfaces. The rotating surfaces revolve freely with ample clearances.

In turbo-compressors manufactured by Pokorny and Wittekind, Brown-Boveri & Company, Esches, Wyss, Sulzer Brothers, Rateau and others, the machines are made up in multiple cylinders according to volume and pressures. The general principle consists of high-speed impellers incased in a cylindrical chamber, properly cooled radially and axially. These impellers draw in and discharge the air the same as a centrifugal pump impeller, turning the velocity into potential energy in the form of air pressure.

It does not appear necessary to use six stages arranged in series for so low a pressure as 10 to 20 lb., and if the compressor end were divided into separate cylinders or casings, each cylinder or casing containing one or more impellers, and the diameter of the impeller and its casing were reduced in the second or third, the number of stages required would be less than that used by the author, provided also that proper cooling was effected radially as well as axially.

A turbo-blowing engine built on the Rateau type for 12,300 cu. ft. of free air per minute down to 6000 cu. ft. for a pressure of from 8 to 13 lb., speed 3400 to 3900, requires three stages only as against six stages of the one described in the paper. This Rateau multi-blower has single entrance impellers, leading the discharge from one wheel into the eye of the other through special channel castings. For pressures of 80 to 150 lb. it is not necessary, if it is properly con-

structed, to use more than 10 to 15 stages, and for lower blowing pressures, 10 to 20 lb., 3 stages, and still maintain an isothermal efficiency of 70 per cent. Such compressors will have a weight about one-quarter of that of a reciprocating outfit.

It has been shown through their use in Europe that turbo-compressors with capacities of 6000 cu. ft. per min. and above have from 70 to 80 per cent efficiency, and those of from 3000 to 6000 cu. ft. per min., 65 to 70 per cent efficiency, and that these efficiencies decrease below 50 per cent at $\frac{1}{2}$ to $\frac{1}{4}$ load. The application of turbo-compressors should be for volumes from 3000 cu. ft. per min. and up.

It is usual to fit compressors with an automatic outlet valve in order to prevent any detrimental pulsation of air at partial capacities, so that the capacity can be kept above the minimum. To prevent an increase of the capacity over the maximum, an automatic valve is installed in the suction.

The use of a turbo-blower in connection with existing reciprocating compressors has been carried out lately in England. The existing compressor was coupled up to a steam turbine turbo-blower, whereby the low-pressure steam cylinder of the reciprocating air compressor exhausted into the steam turbine, and the turbo-blower took the air at atmospheric pressure, compressing it and discharging it into the reciprocating compressor cylinders where it is compressed to 60 lb. pressure.

This combination of turbo and piston-compressor doubled the capacity and gave a net gain of 17 per cent over that which would have been secured had an additional reciprocating compressor been put in. In addition to this it saved the extra cost of a new building on account of the small space it occupied and increased the over-all efficiency, as it utilized the exhaust steam.

It may not be out of order to add the following as characteristics of turbo-blowers and compressors:

a Steady non-pulsating currents when fitted with proper control apparatus.

b Smaller weight and space required, reduction of foundations and handling.

c Absence of reciprocating parts, valves and packings.

d Smaller attention and maintenance.

e Reduction of oil consumption and supply.

f Continuous and steady delivery of air, flexibility of operation, automatic arrangement.

g Utilization of exhaust steam from any reciprocating engine to run the low-pressure turbine operating the blower or compressor.

h Efficiency of the machine, comparing favorably with the best reciprocating compressors.

i Adaptability for cupolas, brass furnaces, converter, operating pneumatic tools, etc.

j Adaptability to operation and furnishing of illuminating gas to mains.

JULIAN KENNEDY[1] said he did not know much either about gas engines or turbines, but he had been doing a little figuring. On a basis of $116 per kw. for gas-engine installations and $60 for steam-turbine installations, he assumed that 20 per cent was a fair amortization charge for up-keep and capitalization for the gas machinery, and 15 per cent for the steam machinery: the investment cost, therefore, for gas machinery would be $23.30 per year, and for the steam machinery $9 per year.

Assuming 7200 hours per year and 2 lb. of coal per kw-hr. at $1.80 per net ton, the cost of coal per kw-hr. would be $0.18, or less than the additional charge of up-keep and capitalization for the gas-engine installation. In other words, according to his figures, if furnace gas were obtained for nothing, and if coal was $1.80 a ton, the result at the end of the year would come out about even.

In his figures he put in furnace gas free, so that if the engine could be brought to such a point of economy as to run without gas, his calculation would hold good provided it cost 5 per cent more to keep up the gas engines than the steam turbines; but he thought 15 per cent would be getting nearer to it.

JOSEPH MORGAN called attention to the fact that the value of the surplus gas and the power generated from it depend entirely on how much can be sold. If there is no market for the power, the gas is worth nothing, while the value of a coal pile remains until used.

C. J. BACON and H. J. FREYN. An analysis (Table 2) of the financial considerations involved in a comparison of a turbo-blower equipment with a gas blowing-engine equipment for blowing a blast-furnace plant consisting of four 500-ton furnaces, leads to results indicating that the total expense for fuel, operation and fixed charges

[1] Consulting Engineer, Pittsburgh, Pa.

is less for a turbo-blower installation than for a slow-speed gas blowing-engine installation in localities where the price of coal establishes a value lower than approximately $0.08 per 1,000,000 B.t.u. in the blast-furnace gas; and is less for turbo-blowers than for high-speed gas blowing engines where the price of fuel is lower than approximately $0.05 per 1,000,000 B.t.u. This conclusion is based on normal conditions prevailing in the average blast-furnace plant, but is influenced considerably by blast pressure, efficiencies, price of labor, water supply for condensers, value set on gas, cost of machinery and construc-

TABLE 2 TOTAL EXPENSE INCLUDING FUEL, OPERATION AND FIXED CHARGES

Cost of Fuel, Cents per 1,000,000 B.t.u.	Turbo-Blowers		Gas Blowing Engines			
			Slow-speed		High-speed	
	Yearly	Per 1,000,000 Cu. Ft.	Yearly	Per 1,000,000 Cu. Ft.	Yearly	Per 1,000,000 Cu. Ft.
			15 lb. Blast Pressure			
4	276700	3.70	340400	4.49	292460	3.85
6	324600	4.34	360000	4.75	312060	4.11
8	372100	4.97	379500	5.00	331560	4.37
10	420100	5.61	399100	5.26	351160	4.63
12	468100	6.25	418700	5.52	370760	4.88
14	516100	6.90	438200	5.77	390260	5.14
			20 lb. Blast Pressure			
4	300100	4.06	350100	4.61	302160	3.98
6	359100	4.80	374500	4.94	326560	4.30
8	419100	5.60	398900	5.26	350960	4.62
10	478100	6.39	423300	5.58	375360	4.94
12	537100	7.17	447700	5.90	399760	5.27
14	597100	7.98	472100	6.22	424160	5.59

tion material, rate of interest, taxes, insurance, estimates of life of apparatus, method of computing depreciation, etc.

For blowing four 500-ton furnaces there would be required eight gas blowing engines, including two spare engines, each with sufficient capacity for supplying two-thirds of the air required per furnace at a normal pressure between 15 and 20 lb. gage and occasionally as high as 25 lb. The usual displacement for a 500-ton furnace is 40,000 cu. ft. per min., although the amount of air actually discharged is somewhat below this figure and is considered as 89 per cent or 35,600 cu. ft. on the basis of actual measurements. Slow-speed engines are supposed

to have a displacement of approximately 500 to 600 cu. ft., and high-speed engines approximately 360 cu. ft. In the case of the turbo-blower installation, six units would be required, including two spare units, each of 35,600 cu. ft. per min. actual discharge capacity at the same pressures. A convenient basis for comparing the costs of operation of different installations is per 1,000,000 cu. ft., in which connection each of the plants under consideration has a total yearly capacity of 74,845 million cu. ft. at 35,600 cu. ft. per min. per furnace.

In preparing the above figures it is assumed that the entire plant of four furnaces would operate continuously at the rated capacity of 500 tons per day. In case of reduced output or of partial shutdown due to dull business, relining furnaces, etc., the reduction in the item of fuel expense would be about twice as great with turbo-blowers as with the gas engines, while the item of fixed charges would continue at the same rate, resulting in a greater net decrease in the total expense for the turbo-blowers than for the gas engines.

Moreover, as certain approximately equal percentage allowances are made in the various efficiencies leading to a comparison of the heat consumption to take account of the actual operating conditions over long periods, any change in the fuel cost due to over-estimating such allowances would also result in a greater net decrease of total expense of the turbo-blowers than the gas engines.

Fuel Expense. Blast-furnace gas, after being subjected to preliminary cleaning, is burned under boilers for furnishing steam power for the turbo-blowers which are direct connected to high-pressure steam turbines. For the gas blowing engines the gas is given both primary and secondary cleaning. In each case the same value is set on the B.t.u. contained in the raw or uncleaned gas according to the local price of coal. This is an equitable basis since a furnace plant of this size would be in close proximity to steel mills in which the power produced from surplus blast-furnace gas is entirely used, thereby displacing coal which otherwise would be burned. The gas is therefore not a waste product.

Since the B.t.u. per lb. of coal, as well as the price, vary greatly in different localities, a unit of cost of fuel per 1,000,000 B.t.u. is used to facilitate comparisons. In the Pittsburgh district the cost is between $0.035 and $0.05 per 1,000,000 B.t.u.; in Ohio, $0.05 to $0.065; in Illinois and Indiana about $0.08.

The heat consumption for blowing the furnace plant under consideration involves a comparison of the various efficiencies affecting the

conversion of the heat in the gas into power as represented by the actual delivery of compressed air. In assigning values, the results of actual performance under operating conditions are used in all cases where applicable data are available (Table 3).

TABLE 3 DATA AND METHOD USED IN OBTAINING THE FUEL EXPENSE

Type of Equipment	Turbo-Blower		Gas Blowing Engine	
Blast pressure, lb. per sq. in.	15	20	15	20
M.e.p. of adiabatic compression, lb. per sq. in.	11.4	14.2	11.4	14.2
M.e.p. in air tubs of blowing engine, lb. per sq. in.	.....	.....	12.1	15.1
Efficiency of compression, per cent	.....	.....	94	94
Displacement of tubs per furnace, cu. ft. per min.	.....	.....	40000	40000
Volumetric efficiency, per cent	.....	.....	89	89
Air delivered to blast mains, cu. ft. per min.	35600	35600	35600	35600
Theoretical air, h.p. per furnace	1775	2210	1775	2210
Air h.p. of blowing engine per furnace	.....	.....	2115	2670
Mechanical efficiency of tub, per cent	.....	.....	92	92
Shaft efficiency, per cent	70	70	77	77
B.h.p. per furnace	2536	3156	2304	2869
Steam consumption, lb. per h.p-hr.	12.5	12.5	.....	.....
Thermal efficiency at shaft, per cent	9.46	9.46	21	21
Thermal efficiency at theoretical air h.p., per cent	6.62	6.62	16.2	16.2
Heat consumed per furnace, 1,000,000 B.t.u. per hr.	68.2	84.8	27.9	34.8
Yearly cost of fuel per 1,000,000 B.t.u for four furnaces				
At $0.04	$ 95600	$119000	$ 39100	$ 48800
At 0.06	143500	178000	58700	73200
At 0.08	191000	238000	78200	97600
At 0.10	293000	297000	97800	122000
At 0.12	287000	356000	117400	146400
At 0.14	335000	416000	136900	170800
Cost of fuel per 1,000,000 B.t.u. per 1,000,000 cu. ft. air				
At $0.04	$1.28	$1.59	$0.52	$0.65
At 0.06	1.92	2.38	0.78	0.95
At 0.08	2.55	3.18	1.05	1.30
At 0.10	3.91	3.97	1.31	1.63
At 0.12	3.83	4.75	1.57	1.96
At 0.14	4.47	5.56	1.83	2.28

Notes on the Turbo-Blower. The shaft efficiency, or the relation of the power in the discharged air to the shaft power for the particular turbo-blower under discussion is shown to be 70 per cent. Recent developments in larger sizes, it is claimed however, have resulted in as high as 72.5 per cent, but as there is no assurance as yet that the efficiency can be continuously maintained at that point under actual operating conditions, 70 per cent is taken as a fair figure for this comparison.

The steam consumption is taken at 12.5 lb. per b.h.p-hr. At 135 lb. gage pressure, 125 deg. superheat and 28.5 in. vacuum and an efficiency referred to the Rankine cycle of 60 per cent, which may be

expected from a turbine of this size, the steam consumption would be 11.9 lb., which is increased to 12.5 due to the effect of variation of operating conditions.

A boiler efficiency of 62.5 per cent is the actual performance shown by numerous tests of gas-fired boilers with superheaters operating under the normal conditions prevailing at a blast-furnace plant. It is estimated that 8 per cent of the total steam generated goes to auxiliaries of boilers and turbo-blowers and to miscellaneous losses. The boilers for the turbo-blowers are considered as operating at 140 lb. gage pressure, 150 deg. superheat and 70 deg. hot-well temperature. Under these conditions, the thermal efficiency of the turbo-blower installation complete would be 9.46 per cent at b.h.p., or 6.62 per cent at h.p. of adiabatic compression.

Notes on the Gas Blowing Engine. The shaft efficiency, or the relation of the power represented by the actual volume of compressed air to the power required at the shaft, consists of compression, volumetric and mechanical efficiencies.

The compression efficiency is the ratio of the work required adiabatically to the work actually done in the blowing tub. The losses consist principally of those due to inertia and mechanical friction of the discharge valves, to the friction of air through the passages and to late opening of suction valves when such occurs. These sources of loss, taken together, amount to an increase in work of 15 to 18 per cent over theoretical adiabatic compression for certain types of pot valves which require considerable excess pressure for opening. The thin plate automatic valves on Slick tubs show very small losses; in fact, tests made on Snow engines show less actual work than theoretical adiabatic compression. Radiation and water-cooled heads have a varying effect on the power requirement. The results of tests on a number of different kinds of blowing tubs show an average loss of about 6 per cent or an efficiency of compression of 94 per cent for the usual blast pressure of 15 to 20 lb., but varying from 82 to 94.6 per cent within the range of 5 to 30 lb. pressure.

The volumetric efficiency is the ratio of the measured volume of air actually delivered (reduced to atmospheric pressure) to the volume of piston displacement. The losses consist of: (*a*) those shown on indicator cards due to the effect of clearance and friction of air through suction valves, which usually range between 0 and 5 per cent; (*b*) leakage of air past the suction valves, discharge valves and piston; (*c*) the effect of preheating the incoming air during the suction stroke by contact with the surface left hot by the preceding discharge stroke. Tests made on seven blowing engines, part with Southwark tubs and

part with Allis-Chalmers tubs having Reynolds valves, show volumetric efficiencies varying from 79.2 to 89.4 per cent, with blast pressure ranging from 10 to 20 lb. approximately. As some of the engines tested had been in service a number of years and were consequently not of recent design nor in first-class condition, it is not reasonable to consider the results obtained therefrom in establishing a rational figure for tubs of modern design and in fair condition. Volumetric efficiencies of 85 to 98.4 per cent were obtained from Allis-Chalmers tubs with Reynolds pot discharge valves, and as this engine had been in service between two and three years, it may be considered in fair condition. Consequently, in this discussion 89 per cent is taken applying at 15 lb. blast pressure, but varying from 91 to 86 per cent at 5 and 30 lb. blast pressure respectively.

The mechanical efficiency of the air tub, or the ratio of the work shown on the air card to the work required at the shaft, ranges from 86.8 per cent at 5 lb. pressure to 95.8 per cent at 30 lb. pressure. Tests on the Snow engines with Slick tubs show 500 h.p. friction loss between the gas and air cylinders, when blowing 35,400 cu. ft. displacement, or 60 r.p.m. at 15 to 20 lb. blast pressure. The actual horsepower developed in the gas cylinders was 2400, while 1900 was shown in the air cylinders, or 500 h.p. loss. As the speed during the tests was greater than necessary for blowing two-thirds of the requirement of a blast furnace, the friction is taken at 420 h.p., the actual loss shown at the required speed. Taking 420 h.p. as the average total friction loss between gas and air cylinders, the corresponding mechanical efficiency is 79 per cent at 1980 i.h.p. which the Snow tests showed necessary in the gas cylinders for 45 r.p.m., or 26,670 cu.ft. displacement at 20 lb. blast. Records of tests on electric engines of similar size and load show about 84 per cent efficiency between gas cylinders and shaft. The difference between 79 per cent and 84 per cent is 100 h.p. chargeable to air tub friction. The remaining 320 h.p. is the friction loss between gas cylinders and shaft. Therefore the shaft horsepower is 1980−320, or 1660. The friction loss of the tubs is $\frac{100}{1660}$, or 6 per cent, and the mechanical efficiency is 94 per cent. This applies to Slick tubs having a minimum of valve gear. For the average of various types of tubs the mechanical efficiency is taken at 92 per cent. These three efficiencies of the air tub taken together show a net shaft efficiency of about 77 per cent within the range of 15 to 20 lb. blast pressure.

The thermal efficiency at the shaft is about 21 per cent for normal operation at a blast pressure between 15 and 20 lb., which is based on the results obtained from tests on Snow engines. Two series of tests

were made, one at constant pressure of 20 lb. with variable speed, and the other at constant speed of 60 r.p.m. with variable pressure. The former showed a thermal efficiency of 30 per cent referred to the i.h.p. of the gas cylinders at 40 to 65 r.p.m. and the latter showed 29 per cent on the same basis at 10 to 20 lb. blast pressure. These efficiencies referred to the gas cylinder power are equivalent to 24 or 25 per cent thermal efficiency referred to shaft power. These tests were made at higher loads than necessary for blowing two-thirds of the requirement of a blast furnace, at which load 23 and 24 per cent were obtained. Since the results were secured during comparatively short test periods of four hours each and since there is no assurance that they would be duplicated in long periods, 21 per cent thermal efficiency is considered a reasonable figure for the purpose of this discussion.

The net thermal efficiency referred to the theoretical air horsepower is the product of the shaft efficiency and the thermal efficiency at the shaft. This amounts to 16.2 per cent at 15 to 20 lb. blast pressure and varies from 9.5 per cent at 5 lb. pressure to 17.8 per cent at 30 lb. pressure.

Operation Expense. As in the case of cost of installation it has been necessary to estimate the costs of the items making up the total cost of operating the turbo-blower equipment. The basis for estimating the gas-cleaning and boiler-plant expense is the actual cost of similar operation under steel-mill conditions and the expenses applying to the turbo-blowers are based on detailed analysis of the various operating costs of a number of high-grade turbo-electric plants of the same size approximately, which seem to be proper sources of data on account of the similarity of equipments. In the case of gas blowing engines the operation expense is the direct result of comparing the actual costs for several gas blowing plants.

COST OF OPERATION (WITHOUT FUEL) OF TURBO-BLOWER INSTALLATION FOR FOUR 500-TON BLAST FURNACES

	Per Year	Per 1,000,000 Cu. Ft. Delivered (74,845 million)
Gas cleaning	$ 8,500	$0.113
Boiler plant expenses	24,000	0.321
Engineers, wipers, etc.	14,600	0.199
Labor and material in repairs and maintenance	10,000	0.134
Lubricants	750	0.010
Tools and miscellaneous supplies	750	0.010
Water	2,500	0.033
Total	$61,100	$0.820

COST OF OPERATION (WITHOUT FUEL) OF GAS BLOWING ENGINE INSTALLATION FOR FOUR 500-TON BLAST FURNACES

	Per Year	Per 1,000,000 Cu. Ft. Delivered (74,845 million)
Cost of purification	$17,600	$0.235
Engineers and wipers	21,100	0.282
Labor and material in repairs and maintenance	20,000	0.268
Lubricants	8,200	0.110
Tools and miscellaneous supplies	850	0.010
Water	4,400	0.059
Electric power	350	0.005
Total	$72,500	$0.970

Fixed Charges. Since there are no turbo-blower installations of this particular size and equipment, the cost of installation has been estimated with due consideration of available data on actual costs of material and construction of similar apparatus. In the case of the slow-speed gas blowing engines, the cost is an estimate on a model installation, combining the best features of a number of actual plants and making use of data on the actual expenditures for plants of capacities almost identical with the turbo-blower installation. Although there are at hand no data on the costs of exclusively high-speed, gas blowing-engine installations, use has been made of the afore-mentioned data on slow-speed equipments modified to take account of reduced space and present prices of the high-speed equipment.

COST OF INSTALLATION AND FIXED CHARGES ON A TURBO-BLOWER EQUIPMENT COMPLETE FOR FOUR 500-TON BLAST FURNACES

	Cost		Estimated Life, Years	Depreciation at 6 Per Cent Interest	Sinking Fund Per Year	
Gas-cleaning plant		$27,000	15	4.2		$1,130
Boiler house						
Excavating and filling	$1,800		..	...	...	
Foundations	2,520		20	2.8	70	
Building	30,600		20	2.8	860	
Floors, etc.	1,080		20	2.8	30	
Total		36,000		2.8		960

COST OF INSTALLATION AND FIXED CHARGES ON A TURBO-BLOWER EQUIPMENT COMPLETE FOR FOUR 500-TON BLAST FURNACES—*Continued*

	Cost		Estimated Life, Years	Depreciation at 6 Per Cent Interest	Sinking Fund Per Year	
Boilers						
Excavating and filling	1,080		..	...	..	
Foundations	5,400		18	3.3	180	
Boilers complete	100,800		18	3.3	3,320	
Steam pipes	28,800		18	3.3	950	
Feedwater system	18,000		15	4.2	760	
Blow-off pipes, etc	4,320		15	4.2	180	
Gas flues, burners, etc	25,200		10	7.2	1,810	
Superheaters	32,400		10	7.2	2,330	
Total		216,000		4.4		9,530
Building for turbo-blower						
Excavating and filling	2,500		..	...	...	
Foundations	4,000		20	2.8	110	
Building	26,000		20	2.8	730	
Floors, etc	8,000		15	4.2	340	
Crane	7,500		15	4.2	320	
Total		48,000		3.1		1,500
Turbo-blowers						
Excavating and filling	5,000		..	...	...	
Foundations	10,000		15	4.2	420	
Turbo-blower erected	320,000		10	7.2	23,000	
Condenser	120,000		10	7.2	8,640	
Piping and sewers	45,000		10	7.2	3,240	
Blast pipes	48,000		18	3.3	1,580	
Total		548,000		6.7		36,880
Total investment		875,000				

Fixed charges	Per Cent	Per Year
Depreciation	5.7	50,000
Interest on investment	6.0	52,500
Taxes	1.5	13,120
Insurance	0.5	4,380
Total	13.7	120,000
Total per 1,000,000 cu. ft. air		1.60

COST OF INSTALLATION AND FIXED CHARGES FOR A SLOW-SPEED GAS BLOWING ENGINE EQUIPMENT COMPLETE FOR FOUR 500-TON BLAST FURNACES

	Cost		Estimated Life, Years	Depreciation at 6 Per Cent Interest	Sinking Fund Per Year	
Gas-cleaning plant						
Excavating and filling	$2,400		...	...	...	
Foundations	9,000		10	7.2	650	
Building	7,000		15	4.2	300	
Gas washers (static)	37,000		15	4.2	1,550	
Gas holder	8,200		15	4.2	340	
Machinery (rotary washer)	23,200		10	7.2	1,670	
Piping	29,200		15	4.2	1,230	
Total		116,000		4.9		5,740
Building for engines						
Excavating and filling	7,000		...	...	...	
Foundations	14,000		20	2.8	390	
Building	122,000		20	2.8	3,420	
Floors	34,000		15	4.2	1,430	
Crane	12,000		15	4.2	500	
Total		189,000		3.0		5,740
Gas blowing engines						
Excavating and filling	14,000		...	...	...	
Foundations	60,000		15	4.2	2,520	
Engines	1,096,000		10	7.2	78,910	
Piping	60,000		10	7.2	4,320	
Blast pipes	75,000		18	3.3	2,770	
Total		1,305,000		6.8		88,520
Total investment		1,610,000				

Fixed charges	Per Cent	Per Year
Depreciation	6.2	100,000
Interest	6.0	96,600
Taxes	1.5	24,150
Insurance	0.5	8,050
Total	14.2	228,800
Total per 1,000,000 cu. ft. air		3.06

COST OF INSTALLATION AND FIXED CHARGES FOR A HIGH-SPEED GAS BLOWING ENGINE EQUIPMENT COMPLETE FOR FOUR 500-TON BLAST FURNACES

	Cost		Estimate Life, Years	Depreciation at 6 per Cent Interest	Sinking Fund Per Year	
Gas-cleaning plant						
Excavating and filling	$2,400		..	...	...	
Foundations	9,000		10	7.2	650	
Building	7,000		15	4.2	300	
Gas washers (static)	37,000		15	4.2	1,550	
Gas holder	8,200		15	4.2	340	
Machinery (rotary washers)	23,200		10	7.2	1,670	
Piping	29,200		15	4.2	1.230	
Total		116,000		4 9		5,740
Building for engines						
Excavating and filling	5,000		..	...	...	
Foundations	10,000		20	2.8	280	
Building	93,500		20	2.8	2,620	
Floors	26,000		15	4.2	1,090	
Crane	9,500		15	4.2	400	
Total		144,000		3.1		4,390
Gas blowing engines						
Excavating and filling	10,000		..	...	....	
Foundations	40,000		15	4.2	1,680	
Engines	842,000		10	7.2	60,620	
Piping	55,000		10	7.2	3,960	
Blast pipes	70,000		18	3.3	2,310	
Total		1,017,000				68,570
Total investment		1,277,000				
				Per Cent		Per Year
Fixed charges						
Depreciation				6.2		78,700
Interest				6.0		76,620
Taxes				1.5		19,155
Insurance				0.5		6,385
Total				14.2		180,860
Total per 1,000,000 cu. ft. air						2.41

Notes on Turbo-Blower Installation. The gas-cleaning plant is composed of static wet cleaners of the tower type with a capacity of 3,300,000 cu. ft. per hr. cleaned to 0.25 grain per cu. ft.

The boiler plant consists of 7200 h.p. of water tube gas-fired boilers with provision for coal firing, superheaters, feedwater system, steam and water piping contained in an all-steel building of the class common in steel plants. The normal requirement of the turbo-blowers including auxiliaries and losses is 6000 b.h.p., the remaining 1200 h.p. being for spare use and cleaning.

The turbo-blower plant includes six turbo-blower units driven by direct-connected high-pressure steam turbines, each having a rated capacity of 35,600 cu. ft. per min. at 15 to 20 lb. blast pressure, there being two units to spare. Each blower has a surface condenser for 40,000 lb. of steam per hour and 28.5 in. vacuum with 70 deg. water. Each condenser requires 7200 gal. of circulating water, pumped by an 18-in. centrifugal pump for each unit. The circulating and hot-well pumps are driven by steam turbines and the air pump by a reciprocating engine. All auxiliaries are located in the basement. The water supply to the pumps is through a 7-ft. conduit, assumed 400 ft. long, and the sewer is of similar size and length. The blast piping includes receivers, valves, cross-connections and all piping to the mains outside the building. The building is 50 ft. by 175 ft. of brick, painted inside, with concrete floor.

Notes on Gas Blowing Engine Installation. The gas-cleaning plant consists of a preliminary and a secondary part, the former comprising three wet scrubbers (tower washers) and the latter three Theisen gas washers operated by electric motors. The mechanical equipment (rotary washers) is installed in a steel building with brick walls, painted inside and provided with all auxiliaries such as pumps, switchboard, wiring, gages, safety appliances, etc. A small gas holder acting as gas-pressure regulator is included, as well as a settling tank to purify the waste water from the gas-cleaning plant and all the piping connecting the various parts of it also provided. The total capacity would be about 25,000 cu. ft. of gas per minute.

The blowing-engine house is a steel frame building with brick walls, painted inside, large windows, ventilation, a skylight, reinforced concrete floors, a spacious basement 10 ft. high, 30-ton electric traveling crane with main and auxiliary hoist, safety and sanitary appliances, electric lighting, office, etc. The width of the building for slow-speed blowing engines is about 105 ft. between the center line of columns and it is about 90 ft. for high-speed gas blowing engines.

The gas blowing-engine equipment consists in the case of slow-speed, gas blowing engines of eight double-acting, twin-tandem, gas engines about 42 in. by 60 in. stroke, operating blowing tubs of about 72 in. in diameter at a normal speed of 50 to 60 r.p.m. The alternative case of high-speed engine equipment comprises eight double-acting, twin-tandem, gas blowing engines; about 38 in. in diameter of gas cylinders; 64 in. in diameter of blowing tubs, 50 in. stroke, operating normally at about 85 to 90 r.p.m.

Depreciation is understood as the amount of money set aside yearly, which compounded at an accepted rate of interest would equal the original investment at such a time as replacement is necessitated through obsolescence, age or inadequacy.

From these figures it will be seen that while a turbo-blower installation under certain conditions is more economical from a commercial standpoint than an installation of slow-speed gas blowing engines, modern high-speed gas blowing engines, as advocated by Mr. Trinks[1] and as used in German blast furnaces and steel plants, offer marked advantages over the turbo-blower as a financial investment.

The present tendency towards high-speed machinery is sure to extend its influence to the gas engine and particularly the gas blowing engine. The only difficulty which so far has stood in the way in this country of the development of high-speed gas blowing engines is the lack of a blowing tub which would operate successfully on high speeds. This side of the proposition has been very ably and fully dealt with in Mr. Trinks' paper[1].

A recent trip through Europe has revealed the fact that in spite of the development of the turbo-blower by several manufacturing concerns in Germany, the gas blowing engine is more than holding its own, due to the high price of coal, since certain turbo-blower installations did not give as satisfactory results as had been claimed for this type of blowing engine.

Looking at the proposition merely from a short-range commercial standpoint, it must be conceded that for isolated blast-furnace plants where no use can be made of the excess gas produced by the blast furnaces, the turbo-blower is unquestionably very desirable, in fact, unsurpassed as an ideal apparatus for blowing smaller furnaces, and it is likely to prevent very successfully the introduction of gas blowing engines in such cases, except when blast-furnace proprietors in a district reach such an agreement as exists in several instances in

[1] Reciprocating Blast-Furnace Blowing Engines, W. Trinks, The Journal, Am. Soc. M. E., July 1911, p. 751.

Germany, for the purpose of producing cheap electric power for sale to communities and industrial establishments of all kinds located in the neighborhood of these isolated plants.

Thus, even in isolated blast-furnace plants very careful investigations should be made before the gas engine is discarded as a possible competitor for steam equipment, since a lucrative field for disposal of surplus electric power may be thrown open by the organization of such syndicates of otherwise competing blast-furnace and steel works. They would have the additional advantage that each constituent company could obtain power from the common system at low cost in case their own power production is curtailed by accidents to their machinery or by reduced output of their furnaces.

In spite of all computations endeavoring to prove that the steam turbine or turbine-driven turbo-blower are commercially superior on account of the low cost of installation and operation, it is impossible to get around the fact that the surplus gas of blast furnaces when converted into power in internal-combustion engines, yields over twice the amount of energy possible when the conversion takes place in the roundabout way of steam boiler and reciprocating engine or steam turbine. To emphasize this statement it needs only to be pointed out that of the total gas production of four 500-ton blast furnaces of about 13,000,000 cu. ft. per hr., there remains available for purposes outside the blast-furnace requirements themselves, 3,750,000 cu. ft. per hr. in the case of the turbo-blower installation, while this surplus amounts to 5,670,000 cu. ft. per hr. when gas blowing engines are used. In the former case an electric station of about 10,000 kw. capacity, equipped with steam turbines could be operated without burning additional coal under boilers, whereas in the latter case a gas-engine central station of about 30,000 kw. capacity could be installed and operated.

In view of the natural tendency towards reducing the cost of installation and operation of power plants and of the constant increase in the price of coal, it is very clear that the internal-combustion engine is bound to gain in value as a prime mover in the future. The smoke nuisance, the difficulties of coal shortage, car famine, etc., in the case of steam-power installations are mentioned only incidentally in their influence upon the probable development of gas power.

Recently means have been found to increase still further the efficiency of gas-power plants by recovering the waste heat formerly lost in cooling water and exhaust. Experiences in a large Belgian blast-furnace and steel plant have proved that about 13 per cent more power can be obtained from the same input of heat units by harness-

ing in the form of low-pressure steam a large portion of the waste heat of gas engines.

In power plants of the future, the installation of waste heat recovery appliances will permit of obtaining continuous high thermal efficiency of the gas engines by maintaining uniform high load on these engines, while peak loads will be carried by low-pressure, steam turbo-generators.

The suggestion offered by Mr. Johnson in regard to a combination of low-pressure steam turbo-blowers and reciprocating steam-blowing engines, can be considerably improved and elaborated upon, since it would seem much more economical to install in a new blast-furnace plant several large turbo-blowers, operated by low-pressure steam turbines, running on steam generated by the waste heat recovered from cooling water and exhaust of gas-electric and gas blowing-engines. These turbo-blowers would deliver the total amount of air necessary for all furnaces at low pressure into an air-tight tunnel or header, from which a number of gas blowing engines would be supplied. The latter would compress the blast in a second stage, as it were, to regular blast pressure.

With such a combination the fuel cost for obtaining low-pressure blast would be practically nothing and the cost of operation and fixed charges moderate, on account of the rotary type and reduced cost of installation of the first-stage machinery. Moreover, the first cost and operating expenses as well as fuel cost for the gas blowing engines themselves would be reduced, since a smaller number of smaller gas blowing engines would suffice. The volumetric efficiency of the blowing tubs would be practically 100 per cent, since the air delivered by the turbo-blowers could easily be cooled by means of pipe coils arranged in the air-tight tunnel, through which the engine cooling water would circulate before entering the gas-engine water jackets.

The safety of blast-furnace operation would suffer in no way, since in case of shutdowns of the turbo-blowers, the gas blowing engines would insure delivery of a reduced quantity of air; while in the event of trouble with the gas engines, large quantities of low-pressure air would be available to keep the gas out of the cold blast pipes.

Since conservative estimates show that at least 12 per cent of the power produced in gas engines is available for low-pressure steam turbines in the waste heat, 25 per cent and more of the power necessary for blowing the furnaces could be secured by the utilization of the waste heat from gas blowing engine and gas-electric engine installations.

THE AUTHOR. The question of efficiency and accuracy of control of the rate of flow of air into the furnace, which is one of the vital questions to be considered in connection with turbo-compressors, consists in reality of two parts, (*a*) a knowledge of the exact rate of flow at any particular time, and (*b*) the accuracy with which this rate of flow can be changed. In both cases the turbo-compressor stands alone, and by virtue of the great accuracy with which the rate of flow can be determined, and controlled and varied through as slight or as great ranges as may be desired, will give unprecedented control over the variations of the furnace to the furnace man, and enable him to produce more uniform quality of iron under greater quantity from any given furnace.

The variation per stroke in volume discharged from any given reciprocating compressor when blowing at pressures of 10 or 20 lb. (both of which are common in practice), due to the variations in the quantity of air required to fill the clearance space at these different pressures, is far greater than the variations due to the inaccuracy of the calibration of the constant volume governor scale beam on the centrifugal compressors. In addition the reciprocating blowing engine suffers from variations due to leakages, pulsations in the blast main, and other causes which, when summed up, render uncertain the exact quantity of air flowing into the furnace. Uncertainties do not exist when the centrifugal compressor is used, for the blast-furnace operator is furnished with a means of measuring the air accurately and controlling it with any necessary degree of precision.

The method proposed by Mr. Dreyfus of blowing air to a constant pressure in a turbine-driven compressor and taking constant pressure into a reciprocating blowing engine is not one made necessary by the considerations set forth. The turbo-compressor can be designed with a reasonable number of stages to give pressures required in the normal operation of the blast furnace, and by a very moderate increase in speed it can be made to give, with good efficiency, the maximum pressures required in such service. Therefore it is not necessary to use this type of blowing engine as a booster. Simplification and accuracy of operation are obtained in the highest degree consistent with maximum economy by taking the air from the turbo-compressor directly into the furnace.

The method of operation where the turbo-compressor is used as a booster does not realize the full benefits of the accurate control and uniform flow of air afforded by the turbo-compressor, and it is there-

fore objectionable. It is, however, extremely economical to use steam from reciprocating blowing engines in a mixed pressure turbine driving a turbo-compressor, and several such installations are now in contemplation. In such cases, the best arrangement is for the reciprocating blowing engines to blow directly into one furnace, and the turbo-compressors into another, allowing each apparatus to be governed in the ordinary manner by its own governing mechanism.

The use of dry blast in a turbo-compressor results in increased efficiency, since the air is delivered to the turbo-compressor at a greater density than where it is taken from the atmosphere direct, and this greater density results in the production of the pressure at a lower speed and less consumption of steam.

The description of the centrifugal air compressor developed by the Westinghouse Machine Company and the high efficiencies claimed for this machine are extremely interesting. These latter would be gratifying if realized in practice, but tests of the Great Falls apparatus which have come under the knowledge of the writer indicate that the measurements of efficiency are made on an erroneous basis, and that such machines not provided with discharge vanes, but with rather imperfect types of impellers, are not able to give the efficiency realized by the apparatus which form the subject of the author's paper.

Mr. de Laval called attention to the fact that it is not necessary to use six stages in series for pressures as low as 10 to 20 lb. This is undoubtedly true where higher speed turbines are available for driving such apparatus. At the time the Empire machine was designed such was not the case. Machines for these pressures are now provided with only three stages and attain a higher efficiency because of the higher speed and the smaller losses resulting. Under these circumstances, it is not desirable to reduce the diameter of the second or third impeller as advocated by Mr. de Laval, but only in cases where the pressures to be realized are considerably higher.

The great differences between the Rateau multi-stage compressors and those of the type described by the author reside in the method of balancing end thrust, by Rateau in separate impellers and in the other type by double inlet impellers, and in the governing mechanism.

The arrangement of impellers used by Rateau results in "pumping" of air back and forth between one and the other at light loads. This pumping effect does not exist in the machine described.

The radical difference in the governing mechanism lies in the General Electric governing mechanism being placed on the inlet, and, therefore, subject to far less variations in density, pressure and temperature of air than where the governor is on the discharge.

Professor Trinks mentioned that many of the turbo-blowers abroad were not in operation. The author believes this to be due to the fact that the method of governing such machines is not sufficiently accurate to cause good results, whereas the accuracy of the apparatus described in this paper leaves nothing to be desired.

Mr. de Laval referred to an automatic outlet valve used to prevent detrimental pulsation of air at partial capacities. The same remarks that have been made in connection with the air governor apply to this apparatus. After extensive experiments the author believes it is much better to control the air pulsations by apparatus on the inlet than on the discharge.

It is gratifying to see that Messrs. Freyn and Bacon apparently agree that for installations involving the blowing of less than four furnaces, the turbo-blower is superior to the gas engine, since they do not attempt to study installations involving a smaller number of units. As regards their figures for plants involving blowing four furnaces or more, there are a number of serious objections to be made. In addition, the author feels that if Messrs. Freyn and Bacon had given full credit to the increased value of the output due to the more accurate governing of the turbo-compressor, and the consequently increased quality of the product and its increased amount due to the more uniform operation of the furnace resulting from the employment of the turbo-compressor with this method of governing, their conclusions would have been modified even in the case of the plants they discuss. It is difficult perhaps for furnace men to realize the value of this feature or to give it proper credit.

Referring to the figures for the installation and operating expenses of various plants, it will be noted that the assumption is made that each gas engine shall have a capacity to blow two-thirds of a furnace. Such a method of operation of furnace blowing units is extremely undesirable, since it leads to the imposition of tremendous strains upon the blowing engine, requiring the outlay of a considerably larger investment cost than has been used by these authors in their tables, or resulting in a much larger cost of repair and maintenance than they have used. Nothing approaching the best results in the operation of a blast furnace can be obtained by such a method of operation. The author knows of two cases where in the use of such

a method serious difficulties and abnormal expenses for repairs to blowing engines resulted.

The adequacy of the number of units specified by these authors is questioned in the case of gas engines to provide the necessary amount of spares that the operation of the group of four furnaces can be maintained at full capacity at all times when the condition of the furnace renders such operation possible. Information from all sources as to the continuity of operation of gas engines for blowing blast furnaces confirms the opinion that instead of eight units as specified by Messrs. Freyn and Bacon, ten units would be none too many. If such is the case, the figures for the cost of installation, interest charges, repairs, operating costs, etc., must be considerably raised against the gas blowing engine with turbo-compressors for this purpose. In this case also, if sufficient first cost was put into the blowing engines to secure necessary reliability and to render the above number of units sufficient, the investment cost would have to be largely increased.

Messrs. Freyn and Bacon have assumed that the entire plant of four furnaces would operate continuously at the rated capacity of 500 tons per day. The author believes that such an installation operating continuously for any considerable length of time at the rated capacity has never existed. Messrs. Freyn and Bacon state: "In case of reduced output, or of partial shutdown, due to dull business, relining furnaces, etc., the reduction in the item of fuel expense would be about twice as great with turbo-blowers as with the gas engines, while the item of fixed charges would continue at the same rate, resulting in a greater net decrease in the total expense for the turbo-blowers than for the gas engines." That is to say, if ordinary instead of theoretical operating conditions had been assumed, the case for the turbo-blowers would be substantially improved.

The vital element in the argument of Messrs. Freyn and Bacon for the maintenance of the gas engine is the question of its efficiency of operation. It is to be regretted that it is not stated how the efficiency of the gas engine or reciprocating blowing tubs was ascertained. It is extremely difficult, if not impossible, to measure correctly by any means in common use the discharge of air from a reciprocating blowing apparatus, owing to the variations in pressure met with in the blast mains. This is due to the fact that the air is discharged from the blowing tub during only a fraction of the stroke, causing a pressure wave to proceed through the blast main at every stroke. It may be that the blast main in this particular case was

so long that these fluctuations in pressure were smoothed out and became negligible; but it would be very interesting to have this point cleared up. If the quantity of air discharged was measured by the indicator card it must be obvious that large errors are likely to exist, and this would seriously affect the computations of efficiency. For instance, if a leak exists in the discharge valves the indicator cards will show a considerable amount of work performed while the air delivery is not in proportion to the amount of work so shown. In comparing turbo-compressors with reciprocating blowing apparatus of any type whatever, this fact should be kept clearly in mind. The quantities discharged by the turbo-compressor are accurately measured within 2 or 3 per cent. The quantity of air discharged by a reciprocating machine is not capable of accurate measurements in the ordinary installation, and figures of volumetric displacement as compared with actual delivery are admittedly at variance by considerable and variable percentages.

Messrs. Freyn and Bacon, in discussing the efficiency of turbo-compressors, assume a figure of $62\frac{1}{2}$ per cent for boiler and furnace efficiency, which is perhaps justifiable if the plant is to be operated in the usual blast-furnace manner. It is practicable by the installation of superheaters and fuel economizers to secure with proper supervision 70 per cent efficiency. Of course if this per cent were used in computations, the figures for the turbo-blower installation would be greatly improved.

In discussing the turbo-blower, Messrs. Freyn and Bacon assume an operating steam gage pressure of 135 lb. There is no reason why a modern plant involving the installation of the best apparatus developed for the purpose could not operate at a pressure of 175 lb., but certainly any figures for cost of operation of turbo-blowers ought to be based on a pressure no lower than 150 lb. Furthermore, Messrs. Freyn and Bacon increase the steam consumption from 11.9 lb., which they derive from a consideration of factors, including this low steam pressure, and the use of efficiency of 70 per cent, which is lower than the figure they themselves give for the turbo-compressor ($72\frac{1}{2}$ per cent), and which is correct, to 12.5 lb., or an increase of 5 per cent. Since they have decreased the efficiency as above by $8\frac{1}{8}$ per cent, this makes a total decrease of efficiency of the turbo-blower, to allow for variable conditions, of $8\frac{1}{2}$ per cent. All the curves and tests on turbo-compressors made by the author indicate that this figure is very much too high. The figure given by Messrs. Freyn and Bacon for the performance of turbo-blowers is therefore unnecessarily conservative.

Although the loss by variation of conditions of air pressure and volume is greater in the gas engine than it is in the steam turbine, no allowances for the effect of these variations have been made by Messrs. Freyn and Bacon in discussing the efficiency of the gas engine, while they have been made when discussing the turbo-compressors. If these allowances are eliminated and 70 per cent is used for furnace efficiency the thermal efficiency of the turbo-blower installation complete would be 11.6 per cent at brake horsepower, or 8.1 per cent at horsepower of adiabatic compression, instead of 9.46 and 6.62 given by these writers. These figures would modify considerably the comparison with the gas engine.

The thermal efficiency of 21 per cent at the shaft, which Messrs. Freyn and Bacon give for the gas blowing engine, should be reduced to at least 19 per cent, if the results of the variations met with in practice are taken into account. In all probability, under all the variable conditions of blast-furnace operation this efficiency would be considerably less than 19 per cent. This would also reduce the thermal efficiency at theoretical air horsepower, given by these writers as 16.2, to not more than 14.6.

In regard to the cost of operation and maintenance, it is of course very difficult to get any figures on the actual cost of operation of installations of either type of blowing apparatus, but from available information the author is inclined to question seriously the adequacy of the item of "labor and material in repairs and maintenance of the gas blowing engine at $20,000 per annum," as being in any way sufficient to cover the cost of this item for an installation of 8 or 10 gas blowing engines.

Referring to the statement of these authors, "In spite of all computations endeavoring to prove that the steam turbine or turbine-driven turbo-blower are commercially superior on account of the low cost of installation and operation, it is impossible to get around the fact that the surplus gas of blast furnaces when converted into power in internal-combustion engines yields over twice the amount of energy possible when the conversion takes place in the roundabout way of steam boiler and reciprocating engine, or steam turbine," it should be borne in mind that the output of gas from a blast furnace is variable and irregular in amount. The author believes that Messrs. Freyn and Bacon do not take sufficient account of the irregularity of operation of plants depending on this gas for their motive power, where such plants cannot be provided with additional sources of heat, for instance, by hand-fired boilers. It is true that a gas

engine can use only gas, and if the gas supply is not sufficient, the gas engine must give out a reduced output, or shut down. In the case of a steam plant containing boilers, if the gas supply is insufficient, the capacity can be kept up by hand-firing of coal. In the case of a gas engine, unless an expensive plant of producers is installed, a deficiency in output will be experienced under such conditions. For these reasons, speculations as to the value of the additional gas set free by the use of a more economical blowing engine must be considered in the light of this irregularity, and additional apparatus consisting of steam engines or steam turbines, provided with the necessary boilers, etc., or a plant of gas producers must be installed to supplement the gas-engine plant if regularity of operation is to be expected and regularity of output ensured. Furthermore, it is difficult properly to govern a gas engine to ensure the necessary regularity and frequency of voltage, and it has been found necessary in some cases to install one or more steam turbines in connection with a gas-engine electric plant taking its gas supply from blast furnaces, in order to provide the necessary regulation. The cost of such installations ought to be added to the cost of investment in the gas-engine plant in order to bring the figures up to a fair basis of comparison.

It should also be borne in mind that the present direction of progress in the operation of blast furnaces is towards a reduction in the amount of the fuel burden. For many years blast-furnaces have been using about twice the theoretical quantity of fuel. It is not impossible that in the near future this figure will be greatly reduced, resulting (*a*), in a decrease in the quantity of gas, and (*b*), in a decrease in its quality. This deficiency will have to be made up by hand-fired boilers or coal-fired producers. Considering the comparison between steam and gas units under these conditions it should be borne in mind that the gas producer is an inefficient piece of apparatus and this, with the low combined efficiency of all the conversion processes involved, will militate against it in such a comparison.

Messrs. Freyn and Bacon devote considerable space to the discussion of a combination of a gas-blowing installation and gas-electric station to which are attached several large turbo-blowers operated by low-pressure steam turbines running on steam generated by the waste heat recovered from cooling water and exhaust. The undesirable features of the tandem operation of turbo-blowers and gas blowing engines which these writers advocate has already been dealt

with. The author furthermore believes that they greatly overestimate the quantity of such available heat, which under the most favorable conditions would amount to not more than 10 per cent. Moreover, he cannot see how under such a combination of conditions such as the writers set forth, a volumetric efficiency of anything like 100 per cent could be secured. He is ready to admit, however, that the leakages would be somewhat reduced by reason of the lower range of pressures dealt with by the reciprocating compressor, but the main causes for the low volumetric efficiency would still remain present.

In conclusion, taking figures obtainable by the average purchaser, made up on the basis used by Messrs. Freyn and Bacon, and giving full credit to the turbo-blower for all improvements made in the operation of the furnace, both as regards quantity and quality of output for the average plant, the gas engine will be found to have a very limited field of application, and only in the very largest plants and under the most favorable conditions in regard to fuel cost will it be found desirable to install such apparatus.

FOREIGN REVIEW

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

An Index to Current Articles in Foreign Periodicals, with Abstracts of Some of the More Important

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TOPICAL INDEX OF CURRENT ARTICLES

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Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *a* comparative; *b* descriptive; *c* experimental; *d* historical; *e* mathematical; *f* practical; *g* general; *h* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Aeronautics

LA TURBINE À AIR COMME PROPULSEUR, C. Canovetti. *L'Aérophile*, December 15. 4 pp., 9 figs. *dh*. Airships can be better propelled by an *aerial turbine* than by a propeller with blades, an aerial turbine being practically a propeller enclosed in a casing, and having a large number of blades. The efficiency of such a turbine would be a function of the ratio n of the absolute velocity V of the efflux of the air and its velocity u of translation; the practical value for this ratio is between 0.5 and 0.6; the useful work increases with V, and decreases as n increases, becoming zero for $n = 1$, and passing through a maximum at $n = 0.5$. The author had a 3 h.p. model constructed in 1906 which does not appear to have given satisfactory results.

INDICATEUR AUTOMATIQUE DE DÉRIVE. *La France automobile et aérienne*, December 2. 1 p., 3 figs. *b*. Description of an *apparatus* designed by Poulalion and Armange for *automatically indicating the drifting of an airship*. The principle of the indicator consists in placing a flexible membrane vertically and parallel to the axis of the airship, in such a manner that the membrane will deflect under the influence of the component representing the force of the drift, and in proportion to its magnitude, and registering the deflection mechanically or electrically.

Air Machinery

UNTERSUCHUNGSSTELLE FÜR PRESSLUFTMASCHINEN, K. Wortmann. *Glückauf*, December 9. 1½ pp., 4 figs. *b*. Description of an *Installation for testing compressed air machinery* at the mines of the Zeche Consolidation in Gelsenkirchen.

*UNTERSUCHUNGEN AN VENTILATOREN, A. Johnen. *Zeits. für Elektrotechnik und Maschinenbau*, November 30. 3 pp., 7 figs., 2 curves. Experimental data on *ventilators*.

Fuel

NOUVELLE MÉTHODE D'ESSAI DES CHARBONS DE Mr. le Dr. H. STRACHE. *Journal des usines à gaz*, December 5. *b*. Description of a new *method of testing coal* used at the Technical School in Vienna. Dr. Strache's apparatus consists of a tube of glass having a very high melting point, in which 1 to 2 decigrams (0.22 to 0.44 lb.) of coal are gasified. The tar produced is absorbed by an asbestos plug, and the gas is conducted to a calorimeter.

A certain amount of it is mixed with air, in any proportion, and exploded by an electric spark in an explosion pipette. The heating value is given by direct reading; the weight of the little tube, before and after gasification, gives the weight of the gas escaped, and coke produced; by weighing in a similar manner the asbestos plug the weight of the tar is obtained. What remains in the little tube (*i.e.*, the coke) is then burned in a current of oxygen, and the contents of carbon, hydrogen, and ash are determined. Finally it is easy to determine the percentage of CO_2 in the products of combustion which permits of determining the percentage of carbon in the gas produced. The method is rapid, and is said to give good results.

Das Sauerstoff-Verhältnis und die Wärmekräfte bei Verbrennungs-Vorgängen, *Zeits. für Sauerstoff-und Stickstoff Industrie*, November. 2½ pp., 1 Fig. *dh.* One of a series of articles on the development of, and principles underlying, the better and more economical *methods of using fuels*. A longer abstract of the whole series of articles will be given later.

Machine à extraire la tourbe, système Strenge. *Le Génie civil*, December 2. 1 p., 1 Fig. *b.* Description of the Strenge (German) machine for cutting, mixing, pressing, and spreading *peat*.

Maschinelle Feuerung für Braunkohlenbriketts. *Elektrotechnischer Anzeiger*, December 7. 2 pp., 6 figs. *b.* Discussion of *mechanical stoking for lignite briquettes*, and description of the Pluto stoker, and a so-called self-stoker.

Zentralanlagen zur Beseitigung des Rauches, O. B. *Prometheus*, December 2. 1 p. *g.* Discussion of the plan proposed by Fichtl and Lemberg for the abatement of smoke in cities. The plan consists in conveying all the smoke from residence chimneys and factory stacks by underground passages to one central purifying plant where useful by-products would be separated, and the whole so purified that only oxygen and carbon dioxide would be left free to escape into the air. The author believes in the technical feasibility of the project, but points out the great difficulty of making it pay, at least apart from indirect benefits that might accrue to the community in the form of better health and cleaner air.

Furnaces

Die Anwendung der Gesetze der Hydraulik auf die Berechnung der Flammöfen. *Stahl und Eisen*, December 7 and 22. 15 pp., 7 figs. *h.1.* Exposition of Prof. Grum-Grzimailo's theory of the applicability of the laws of hydraulics to the calculation of *furnaces*, in particular reverberatory.

Neue Heiz-und Schweissbrenner für gasförmige und flüssige Brennstoffe. *Zeits. für Beleuchtungswesen*, November 30 and December 10. 5 pp., 10 figs. *b.* Description of the latest types of *burners for gaseous and fluid fuels*, containing drawings and short extracts from German patents.

Gas Plant Engineering

Die Gewinnung der Nebenerzeugnisse beim Gaserzeugerbetrieb, A.

Gwigger. *Stahl und Eisen*, December 21, 3 pp. *c.* Discussion of the advantages of recovering the useful by-products (*coal-tar and ammonium sulphate*) when operating producer-gas plants. The figures in the article are taken from British practice.

Hydraulics

BARRAGES ET SYPHONS AUTOMATIQUES. *La Technique moderne*, December. 1½ pp., 5 figs. *b.* Description of an *automatic dam* for maintaining the water at a mathematically constant level independent of variations in the supply of water.

TURBINANLAEGGET VED ADAMELLO. *Elektroteknisk Tidsskrift*, December 7. 3 pp., 6 figs. *b.* Description of the *water turbine installation* in Adamello (Italy).

HOCHDRUCK-FREISTRAHLTURBINEN. *Der praktische Maschinen-Konstrukteur*. December 7. 2 pp., 1 sheet of drawings. *h.* Discussion of the *free-jet turbine*, its advantages, principles of construction, and adaptability for high speeds.

L'USINE HYDRO-ÉLECTRIQUE DE BEAUMONT (Izère), M. Gariel. *Le Génie Civil*. December 9. 4 pp., 7 figs. and 1 sheet of drawings. *b.* Description of the *hydro-electric plant* at Beaumont, Izere France, containing drawings of a *turbine speed governor*, Neyret-Brenier system.

COMPTEUR À DISQUE POUR EAU CHAUDE, SYSTÊME SIEMENS ET HALSKE. *Le Génie Civil*, December 2. ½ p., 1 fig. *b.* Description of the Siemens and Halske *hot water meter*.

DIE WASSERDRUCKMOMENTE DER DREHSCHAUFELN VON ZENTRIPETAL FRANCIS-TURBINEN, Prof. R. Camerer. *Zeits. des Vereines deutscher Ingenieure*, December 2. 9 pp., 21 figs. *h.* Description of an experimental and a theoretical method of determining the *moments of water pressure in the wheel-vanes of centripetal (inward flow) Francis turbines*, showing that such moments can be predetermined theoretically with sufficient accuracy.

DIE NEUE HOCHDRUCKTURBINEN- UND PUMPENANLAGE DES ELEKTRIZITÄTSWERKES BERGAMO. *Der praktische Maschinen-Konstrukteur*. December 21. 1 p., 1 table of drawings. *b.* Description of the general installation of the *hydro-electric plant* at Bergamo.

Internal Combustion Engines

*DIE GASTURBINE. *Stahl und Eisen*, December 7. 1½ pp., 1 fig. and 1 indicator diagram. *b.* Extract from a paper read on the *Holzwarth-Junghans gas turbine*, by H. Holzwarth, before the German Society of Naval Construction.

NEUERE ROHÖLMOTOREN, Ch. Pohlmann. *Dinglers Polytechnisches Journal*, December 16. 2 pp., 8 figs. *b.* Description of the *Diesel motor* built by the Aktiebolaget Diesels-Motorer in Stockholm.

D.A.C.-SCHIFFSZWEITAKTMASCHINE. BAUART DIESEL, J. Vollmer. *Die Gasmotorentechnik*, December 16. 7 pp., 10 figs. *b.* Description of a *two-cycle marine motor* of the Deutsche Automobil-Constructionsgesellschaft.

by J. Vollmer, chief engineer of the company. The machine has an interesting system of *cylinder cooling* (air). Separate cylinders with their connections are all made exactly alike, so that the same "cylinder sets" can be used for two, four, etc., cylinder engines.

GROSS-OELMOTOR. *Stahl und Eisen*, December 7. ½ p., 1 fig. *b*. Extract of a paper read by Prof. H. Junkers before the German Society of Naval Construction on the construction of a *large (1000 h.p.) oil motor*. His motor has four plunger pistons which move in pairs in opposite directions in two cylinders. This makes the construction of the cylinders comparatively simple, but requires a somewhat complicated crank mechanism. The author claims for pistons running in opposite directions the following advantages: (*a*) simple valve gear by slots in the cylinder walls and pistons, an effective cooling of the cylinder walls being produced at the same time by the entering air, which allows a higher output for given dimensions of cylinder, and is also of value for the safety and economy of operation of the machine; (*b*) the engine bedplate is not called upon to transmit cylinder pressures; (*c*) reduction of reaction of pressures exerted by the masses to very small amounts.

Machine Shop

FABRICATION ET MODED'EMPLOI DES MEULES D'ÉMERI, F. H. *Le Génie Civil*, December 9. 1½ pp. *c*. Extract from a paper read by W. Hermingshausen at a meeting of German foundrymen, on various substances used for *abrasive wheels*, and methods and cost of using them, giving some practical rules on that subject based on materials obtained from the principal German manufacturers.

WARUM SOLL MAN SEINE GUSZCHARGEN NACH ANALYSEN UND NICHT NACH DEM BRUCHAUSSEHEN DES ROHEISENS GATTIEREN? Pfeiffer. *Giesserei-Zeitung*, December 1. 2 pp. *h*. Second and last of two articles discussing the necessity of determining by analysis the presence and amount *of impurities in cast iron*, and not relying on the appearance of fracture alone.

*TRAITEMENTS THERMIQUES ET MÉCANIQUES DES MÉTAUX À L'ATELIER, F. Robin and P. Gartner. *Revue de mécanique*, November 30. 14 pp., 4 figs. *h*. Discussion of the theory and practice of *annealing* of various kinds of steel, iron, bronze and brass.

DIE HERSTELLUNG DES SCHMIEDBAREN GUSZES (TEMPERGUSZES) IN THEORIE UND PRAXIS, M. Lamla. *Giesserei-Zeitung*, December 1. 3 pp. *A*. One of a series of articles on the production of *malleable cast iron*.

Mechanics

DÉFINITION GÉNÉRALE DE L'ELLIPSE D'ÉLASTICITÉ DES SYSTÈMES ARTICULÉS, B. Mayor. *Bulletin technique de la Suisse Romande*, December 10. 2½ pp. *cA*. The first of a series of articles. The author strives to define a geometric element in such a way that it would coincide with the definition of an *ellipse of elasticity* whenever such an ellipse can exist according to the usual definition, but which would at the same time be free of its restrictions, *i.e.*, not limited to a triangled system free in its plane. The author makes use for this purpose of a special type of trilinear coördinates.

URSACHEN UND BERECHNUNG DER KRITISCHEN TOURENZAHL, Fritz Reuter. *Zeits. für Elektrotechnik und Maschinenbau*, December 7. 2 pp., 4 figs. *e.* Exposition of the nature and method of calculation of the *critical number of revolutions.*

EMBRAYAGES RÉCENTS. *Revue de mécanique*, November 30. 40 pp., 90 figs. *bA.* Description of recent gearing, with a list of more than one hundred articles on *gearing* printed in the *Revue de mécanique* during the last ten years, and covering practically every system that appeared during that time.

TRANSMISSION À DISTANCE DES MOVEMENTS DISCONTINUS SYSTÈME HERZMARK, Ch. Jacquin. *Le Génie Civil*, December 16. 2 pp., 5 figs. *b.* Description of a system of *transmission of intermittent motion* capable of transmitting not only a rectilinear pull, but also a push.

ETUDE CRITIQUE DES DISTRIBUTIONS COMMANDÉES PAR CAMS, F. Carlès. *La Technique automobile et aérienne*, December 15. 3 pp., 6 figs. *h.* Theoretical investigation of *cams* for valve gears.

ETABLISSEMENT DU GRAISSAGE DANS LE MOTEUR ET LA TRANSMISSION, A. Guéret. *La Technique automobile et aérienne.*. 7 pp., 6 figs. *h.* Discussion of the principles underlying correct arrangement of *lubrication.*

Pumps

AUTOMATISCHE PRESZLUFT-SCHWIMMERPUMPE. *Der Bergbau*, November 16. ½ p., 1 fig. *b.* Description of a self-starting pump lately placed on the market in Germany, for use in places which have to be kept dry but where water may appear unexpectedly in not very large amounts. The principle of the pump is as follows: The water runs through an inlet orifice into a float, which sinks by gravity, closing the inlet orifice, and lets in compressed air which forces the water out into the delivery pipe. When the float is emptied it rises, and the operation is repeated.

Refrigerating Machinery

EINE AETHYLCHLORID-KÄLTEMASCHINE, R. Plank. *Zeits. für die gesamte Kälte-Industrie*, December. 1½ pp. Mr. Zimmerman demonstrated before the Society of Oxygen and Nitrogen Industries, November 2, 1911, his *ethylchloride* (C_2H_5Cl) *refrigerating machine.* The machine was very crudely made, but in three tests gave the following results of cold produced per h.p. hour:

	Cal. per German h.p	B.t.u. per American h.p.
Test 1	1,820	8,338
Test 2	1,610	7,370
Test 3	2,555	11,715

The curve of ethylchloride is such that a vacuum must be maintained; it does not attack metals, requires no lubrication, is not poisonous, solidifies at about —100 deg. cent. (—148 deg. fahr.), is not readily absorbed by water, burns, but does not explode; present price 350 M. per 100 kg. (about $0.39 per lb.), but probably will soon be reduced by one-half this amount.

The great objection to the possibility of such machines being built in large sizes is the large cylinder required, the cylinder volumes for different refrigerating substances being in the following ratio: $CO_2 : NH_3 : SO_2 : C_2H_5Cl = 1 : 5.87 : 15.2 : 37$.

DIE 13. HAUPTVERSAMMLUNG DER SCHIFFBAUTECHNISCHEN GESELLSCHAFT, O. Flamm. *Schiffbau*, December 27. 2½ pp. *g*. By a steam jet of high velocity in a jet refrigerating machine designed by Prof. Josse, of Charlottenburg, water vapor is drawn off from a space of very low pressure (about 0.006 kg/qcm, or 0.085 lb. per sq. in.), and evaporation of water produced. This leads to fall of temperature, and Prof. Josse has shown slides of his machine which, without any moving parts, produces a temperature of —17 deg. cent. (+1.4 deg. fahr.).

UEBER DIE BEKÄMPFUNG HOHER TEMPERATUREN IN BERGBAU, Dr. Dietz. *Zeits. für die gesamte Kälte-Industrie*, December. 2 pp. *f*. According to German law, the working day for men in mines where the temperature is higher than 28 deg. cent. (82.4 deg. fahr.) is limited to six hours. It is therefore an important problem to *keep the temperature in deep mines as low as possible*. This cannot be done by ventilation alone because too violent ventilation is apt to increase the danger of explosions, and put out the safety lamps. Installation of steam-driven refrigerating machinery under ground is inconvenient because the refrigerating liquids are of such a nature that, in case of accident, their spread in the mine would be extremely dangerous. To have the machinery above ground would not do either, owing to the difficulty of having vertical piping some 700 m. (2300 ft.) long stay airtight with 80 atmospheres pressure. The author recommends, therefore, the use of a *turbine-driven cold air machine*. It does not appear, however, that such a machine has ever been put on the market.

VERBESSERTE CO_2 EXPANSIONS-KÄLTEMASCHINE, Ludwig Horst. *Zeits. für Sauerstoff und Stickstoff-Industrie*, December. 5 pp., 1 fig., 4 diagrams. *b*. Description of an improved *refrigerating machine using carbon dioxide* in the form of a gas in the expansion cylinder, and discussion of principles underlying the use of CO_2 in refrigerating machinery.

Safety Appliances

INSTALLATIONS DE SÛRETÉ POUR LA MANUTENTION DES LIQUIDES INFLAMMABLES, APPAREILS MARTINI ET HUNEKE, M. François. *Le Génie Civil*, December 9. 4 pp., 9 Figs. *b*. Description of the Martini and Huneke *apparatus for safe storage of inflammable liquids*, particularly hydrocarbons. The inflammable liquid is protected from contact with air by an inert gas under pressure, which also serves to drive the liquid out of the vessel as soon as the cock is opened, the apparatus being so arranged that should a leak occur anywhere, the liquid is relieved of its pressure, and does not flow, thus making an explosion impossible.

Steam Engineering

NOUVELLES MACHINES À VAPEUR. *Bulletin technique de la Suisse Romande*. 3 pp., 12 figs. *a*. Comparison of Stumpf *straight-flow steam engines* with van der Kerchove engines, and attempt to show that the suc-

cess of the first is due not to the use of the principle of straight flow, but to the presence of certain constructive details which were used long ago in the van der Kerchove engines. The data on which the article is based have been published before.

ERGEBNISSE DER ABNAHMEVERSUCHE EINER Z-M.-GLEICHSTROM-DAMPFMASCHINE. Karl Beneke. *Zeits. für Dampfkessel und Maschinenbetrieb*, December 15. 4 pp., 10 figs., 4 tables, of which 3 are data sheets of tests. *bc.* Description of an improved *straight-flow steam engine*, and data of a test of it.

DER LOLLAR-GEGENSTROM-GLIEDERKESSEL. *Der praktische Maschinen-Konstrukteur*, December 21. 2 pp., 7 figs. *b.* Description of the Lollar counter-current sectional *boiler* for central heating. The boiler sections have such a shape that when put together, ducts for the passage of the gases of combustion are formed between them. There is no built-in connection between the boiler and smoke stack, the gases of combustion being simply led off to the stack from the furnace by a smoke tube. All the smoke tubes are placed on one side of the boiler.

NEUERUNGEN IM DAMPFKESSEL UND FEUERUNGSBAU. *Schweizerische Bauzeitung*, November 18. 6 pp., 9 figs. *bc.* Description of *underfed furnaces* manufactured in Switzerland, and data from tests of same.

*LA RÉGULATION D'UNE TURBINE ALIMENTÉE EN VAPEUR À HAUTE ET BASSE PRESSION PAR UN ACCUMULATEUR HARLÉ. *Le Génie Civil*, December 2. ½ p., 1 fig. *b.* Description of the Harlé *accumulator for regulating the supply of steam* to a turbine working simultaneously on exhaust steam from other engines, and live steam from a boiler.

*EINE NEUE UEBERHITZERKONSTRUKTION. *Der praktische Maschinen-Konstrukteur*, December 7. 1 p., 4 figs. *b.* Description of a new *superheater*.

THÉORIE DU TUBE MANOMÉTRIQUE. *Revue de mécanique*, November 30. 7 pp., 4 figs. *c.* Mathematical determination of *the deformation of the tube in a Bourdon manometer*, and its relation to the indication of the manometer needle. The article is based on an earlier article in the *Zeits. des Vereines deutscher Ingenieure*.

SUR LE TIRAGE RATIONNEL DES GÉNÉRATEURS DE VAPEUR. J. Izart. *La Technique moderne*, December. 4 pp., 9 figs. *bh.* Continuation of the article (abstracted in THE JOURNAL last month) on rational *draft for boiler plants*. Description of systems used in France, in particular draft by ejector.

*UEBER METHODEN ZUR VERHÜTUNG UND ENTFERNUNG DES KESSELEINSATZES, W. Dahse. *Zeits. für Dampfkessel und Maschinenbetrieb*, December 8. 1½ pp. Discussion of various methods of preventing the formation and removal of *scale in boilers*.

AUFSTELLUNG BESONDERER MOTOREN FÜR LICHTMASCHINEN. *Elektrotechnik und Maschinenbau*, December 10. *f.* Article based on K. Hauck's article

in the *Zeits. der Dampfkesseluntersuchungs- und Versicherungs-Gesellschaft*, September. The author is opposed to the practice of *driving the dynamo for lighting* a factory from the main steam engine rather than *from a separate engine*. Apart from variations in the intensity of light due to variations in the load on the engine, the author points to steam losses due to underloading such an engine when the lighting circuit is not in use. Two indicator diagrams show that an underloaded engine used 33.4 kg. per h.p-hr., while it ought to have used only 21 kg. per h.p-hr. (73.5 lb. against 46.2 German h.p., or 84 lb. against 55 lb. per American h.p.), involving an increase of cost of operation which would amply pay for an installation of separate engines, one for doing the main work of the factory, and another for driving the lighting dynamo.

CHLORIDE IM KESSELSPEISEWASSER, E. E. Basch. *Technische Rundschau*, No. 46, from extract in *Braunkohle*, December 15. *f. Magnesium chloride in boiler water* is one of the most powerful corrosive agents, attacking the boiler plates especially at the water line where the rise and fall of water afford the time for the deposition and decomposition of salts, and subsequent moistening of the affected surface. To make magnesium chloride harmless, the author recommends the use simultaneously of lime and soda, whereby magnesium chloride is converted into magnesium hydroxide.

OBSERVATIONS ET FORMULES RELATIVES À L'ÉCOULEMENT DES FLUIDES ÉLASTIQUES, Maurice Leblanc. *La Lumière électrique*, December 16. 5 pp., 5 figs. *ch.* Investigation of the flow of elastic fluids in which the author shows that a steam jet escaping from a nozzle at high velocity possesses a sort of individuality of its own, and does not necessarily take the form of the passage through which it is led. The article is to be continued and a fuller abstract will be given next month.

*ABWARMEVERWERTUNG BEI DAMPF-UND VERBRENNUNGSKRAFTMASCHINEN, Dr. Deinlein. *Zeits. des Bayerischen Revisions-Vereines*, October 15, from extract in *Elektrotechnik und Maschinenbau*, December 24. *a.* The author shows by tables that, as regards total *utilization of heat*, the steam and internal-combustion engine are about equal when the waste heat is also utilized. For 100,000 W.E. (397,000 B.t.u.) in fuel put into the engine, a steam engine with 70 per cent boiler efficiency will give 67 to 68.6 per cent total efficiency, and with 80 per cent boiler efficiency 76.5 to 78.1 per cent total efficiency, these figures covering engines working with saturated or superheated steam, non-condensing or condensing. Corresponding values for internal-combustion engines are 75 to 78 per cent. These figures presuppose, however, utilization of all the waste heat, and the author goes on to show that such utilization is really limited. With steam engines steam of any pressure desired, up to that in the boiler, can be taken from the engine, and water heated by it to any temperature up to that of saturated steam, while with internal-combustion engines water can be heated practically only up to 100 deg. cent. (212 deg. fahr.), and even that in small amounts. In general, where heating is required, a steam engine must be considered first, and where it is not required, an internal-combustion engine.

UMLEGUNG EINES DAMPFSCHORNSTEINS. *Tonindustrie-Zeitung*, December 21. Description of *dismantling* a 30-m. (98-ft.) high *smoke-stack* without the use of explosives, the main problem being to make the stack fall in a given direction. Part of the foundation on the side where the stack was required to fall was taken out and carefully replaced by strong wooden props; petroleum oil was then poured over them, and the props set on fire. The area covered by the falling bricks was 35 m. by 9 m. (115 ft. by 29 ft.).

*UNTERSUCHUNGEN ÜBER DAS ALLGEMEINE VERHALTEN DES GESCHWINDIGKEITS-KOEFFIZIENTEN VON LEITVORRICHTUNGEN DES PRAKTISCHEN DAMPFTURBINENBAUES, Paul Christlein. *Zeits. des Vereines deutscher Ingenieure*, December 16. 9 pp., 8 figs., 19 curves. *h.* Investigation of conditions under which flow takes place, and losses occur, in nozzles and distributors of steam turbines now in use, based on experiments made in the mechanical laboratory of the Charlottenburg Technical Institute, with particular regard to the relation between velocity of flow and efficiency.

1500 P. S. DAMPF-GENERATOR VON FRIEDR. KRUPP A. G., GERMANIAWERFT, UND DEN SIEMENS-SCHUCKERT WERKEN, BERLIN. *Zeits. für Elektrotechnik und Maschinenbau*, December 13. 2 pp., 2 figs. and 1 sheet of drawings. *b.* Description of a *Zoelly* 1500-*h.p. turbine* built by the Friedr. Krupp Co.

Strength of Materials and Testing of Materials

LA RÉSISTANCE À L'ÉCRASEMENT DES FERS ACIERS ET ALLIAGES, F. Robin. *Bulletin et comptes rendus mensuels de la Société de l'industrie minérale*, August-December. 425 pp. *b.* Investigation of *resistance to crushing* of steel as a function of temperature.

UEBER EXCENTRISCH GEDRÜCKTE STÄBE UND ÜBER KNICKFESTIGKEIT, Muller-Breslau. *Der Eisenbau*, December. 12 pp. *eA.* Theoretical investigation of the *behavior of bars* under pressure applied eccentrically and of resistance to axial compression.

BEITRAG ZUR BIEGUNGSLEHRE, Otto Mohr. *Der Eisenbau*, December. 5 pp., 10 figs. *chA.* Application of Prof. Mohr's method of solving problems on deformations of elastic bars to such problems as the derivation of the fundamental equation for the calculation of continuous *beams*, pressures at the supports of beams with fixed ends having a polygon or curved cross-section, and stresses and deformations in bars fixed at one end, or having more complicated systems of support.

VERSUCHE MIT ALLUMINIUM GESCHWEISZT UND UNGESCHWEISZT, BEI GEWÖHNLICHER UND HÖHERER TEMPERATUR. *Zeits. des Vereines deutscher Ingenieure*, December 2. 3 pp., 3 figs. *h.* Data on strength of aluminum bars welded, hammered, annealed, etc.

VERSUCHE MIT EINEM AUTOGEN GESCHWEISSTEN BLECHBEHÄLTER AUSGEFÜHRT VON DER FIRMA GEBR. SULZER, WINTERTHUR. *Acetylen*, No. 22. 2½ pp., 7 figs. *b.* Description of tests made by the above-named concern of a plate tank having an *autogenously welded cylinder cover*. Rupture occurred at a pressure of 83 atmospheres.

LES CHEMINS DE FER DU BRÉSIL, Lionel Wiener. *Revue générale des*

chemins de fer, December. 30 pp., 12 figs. *bc.* Description of railway construction in Brazil. Contains data on the *strength of Brazilian stone and timber.*

MACHINE ÉLECTRIQUE POUR L'ESSAI DES MATÉRIAUX. *La Revue électrique*, December 8. Description of an *electric testing machine* invented by Gisbert Kapp for long and often-repeated tests of materials.

APPAREIL SIMPLIFIÉ, SYSTÈME STERN-SONNEBORN, POUR L'ESSAI DES HUILES DE GRAISSAGE. *Echo des mines et de la métallurgie*, December 21. 1 fig. Description of an *apparatus for rapid testing of lubricating oils.* The oil is atomized in air or steam, and the mixture forced into a cylindrical box with two concentric helicoids inside, one going up and the other down. The mixture enters the box through an orifice at the bottom, passes up one helicoid and down another, and finally escapes, leaving in the serpentines of the instrument a deposit which by its appearance and amount shows the quality of the oil.

ERGEBNISSE VON SEILPRÜFUNGEN. *Der praktische Maschinen-Konstrukteur*, December 21. (From the last bulletin of the Royal Testing Laboratory at Gross Lichterfelde West, not yet received in this country). *c.* Data of endurance test of ropes. Ropes were laid over a pulley 5.3 cm. (2.07 in.) in diameter, a load of 20 kg. (44 lb.) placed at one end, and the other end connected with an eccentric in such a way that the rope made 15 movements per minute of 35 cm. (13.8 in.) each. The ropes tested had diameters: 0.5, 0.7, and 1.1 cm. (0.196, 0.275, and 0.42 in.). The ropes broke after from 7,200 to 18,000 movements when the ratio of the diameter of the rope to that of the pulley was 1 : 10, and after from 5,300 to 7,300 movements when this ratio was 1 : 7.5. This shows that the ratio ought not to be chosen too small. As regards the tendency of spiral steel ropes to twist, the amount or torsion in a rope 100 cm. (39.37 in.) long, at a load of 10 tons, or about one-eighth of the breaking load, was found to be 3.0, the torsion and elongation being approximately proportional to the load up to 35 tons, but at higher loads growing (particularly the elongation) more rapidly than the load. Ropes made by a special process out of 32, 120 and 192 piano wires had a strength equal only to about 30 to 55 per cent of the sum of strengths of the composing wires, the ratio being the less favorable the more wires the rope was composed of.

*OPYTY NAD VYRYVANIEM SVIAZEI I VYPOOCHIVANIEM PLOSKIKH STIENOK, M. V. Gololobove. *Bulletin of the Permanent Committee of the Conferences of Representatives of Russian Railroads*, No. 10. 12 pp., 18 figs. *c.* Results of experiments on tearing out of *bolts* and flexure of flat boiler walls, with reference chiefly to locomotive boilers.

Thermodynamics

CONTRIBUTION À LA THERMO-DYNAMIQUE DES GAZ, Charles Beer. *Revue universelle des mines*, No. 3. 48 pp. *eh.* Theoretical investigation in thermodynamics of gases, starting from the following two principles: (*a*) the temperature is proportional to the number of oscillations of molecules per unit time; (*b*) the quantity of heat in a body is represented by the

increase of molecular kinetic energy of the body. The aim of the author is to show the connection between the production of heat and universal gravitation. (An article devoted to the same subject was published by H. C. Marx in *Die Turbine*, March 20, 1911, under the title "Gravitation und Wärme").

Ueber die Molekularwärme des Wasserdampfes, Herbert Levy. *Berichte der deutschen Physikalischen Gesellschaft.* 2 pp. *h.* Notice of a new investigation of the *specific heat of steam* containing a table of some of the new values (very close to the generally accepted ones). The new investigation is claimed to show, among other things, that, as the temperature rises, the isobars of steam tend to converge in a common line.

Der Wärmeübergang im Kreuzstrom, Wilhelm Nuszelt. *Zeits. des Vereines deutscher Ingenieure*, December 2. 3 pp., 5 figs. *e.* Investigation of the *flow of heat in a surface cooler* when the directions of the two streams of liquid are at right angles.

La transmission de la chaleur à travers les parois d'un local chauffé. *La Métallurgie*, December 13. 2/3 p. *hA.* Continuation of a series of articles by E. Mathieu (reprinted from the *Annales des travaux publics de Belgique*) comparing various methods of estimating, and data on, the *transmission of heat through walls and windows of houses.*

Die spezifische Wärme c_p der Luft bei Zimmertemperatur und tiefen Temperaturen, Karl Scheel and Wilhelm Heuse. *Physikalische Zeits.*, December 1. 2 pp., 1 fig. *h.* (Same article in *Berichte der Deutschen Physikalischen Gesellschaft*, No. 21.) The authors have redetermined the specific heat of air by Barnes and Callendar's method of continuous calorimetry, and have found the following values for dry atmospheric air, free of carbon dioxide, under atmospheric pressure:

Temperature (deg. cent.)	Watt-seconds / g. deg.	g. calorie$_{15}$ / g. deg.
+20	1.009	0.240
−78	1.019	0.243
−183	1.058	0.253

Miscellaneous

Niveau de précision.. *Portefeuille économique des machines*, December 2½ pp., 5 figs. *b.* Description of a *level*, Leneven system simplified by H. Morin, for exact levelling, e. g. of shafting. Its chief parts are two vessels connected by a flexible rubber tube; each vessel contains a liquid, and is provided with a little rod, the point of which is brought in contact with the surface of the liquid; a graduated scale (with a vernier) shows the distance between the base of the vessel and the upper surface of the liquid, the difference in reading of the scales in the two vessels indicating the difference in level of the surfaces on which the vessels are standing.

Betrieb und Organisation eigener Patentenbureaus in Maschinenfabriken, Th. Schuchart. *Technik und Wirtschaft*, December. 12 pp., 9 figs. (forms of registration cards, etc.). It has been found in Germany

that it pays even a medium-size machine factory to have its *own patent bureau*. The article describes the installation and methods of work of such a bureau in a fairly large German factory (Thyssen & Co., Mühlheim, Ruhr).

CHARGEMENT AUTOMATIQUE POUR CAMION AUTOMOBILE. *La France automobile et aérienne*, December 2. 2 pp., 4 figs. *b*. Description of a device for *loading an automobile truck* by its own power. The truck is provided with a small jib and a transmission consisting of a toothed segment on the jib and a screw-spindle engaging with it and receiving its power by an ordinary gear transmission from the motor of the truck.

L'OUTILLAGE TECHNIQUE ET PRATIQUE DU DESSINATEUR, J. Escard. *Revue de mécanique*, November 30. 24 pp., 48 figs. *b*. Description of instruments and methods for the optical, mechanical and photographic *reproduction of drawings*, and discussion of the best arrangement of a *drawing-room*.

NEUE ENTSTAUBUNGSEINRICHTUNGEN AUF RHEINISCHEN BRAUNKOHLENBRIKETTFABRIKEN, *G. Polster*. *Braunkohle*, December 22. 7 pp., 6 figs. *b*. Description of various appliances for the *removal of dust* in lignite briquette factories in the Rhine province.

DIE NEUERE ENTWICKLUNG DER FÖRDERMASCHINENANTRIEBE UND DER SICHERHEITSVORRICHTUNGEN, A. Wallichs. *Zeits. des Vereines deutscher Ingenieure*, December 2. 5 pp., 34 figs. *b*. Discussion of the modern types of *winding machines* and safety appliances.

SELBSTSTÄTIGER TEMPERATURREGLER SYSTEM BRABÉE-FUESZ. *Gesundheits-Ingenieur*, December 2. 1½ pp., 6 figs. *b*. Description of the Brabée Fuesz (R. Fuesz in Steglitz b.Berlin, Germany) *temperature regulator* for steam and water heating. The regulation is produced by the change in the volume of a liquid (alcoholic) due to changes in temperature.

*EINHEITSFARBEN ZUR KENNZEICHNUNG VON ROHRLEITUNGEN IN INDUSTRIELLEN BETRIEBEN. *Stahl und Eisen*, November 30. *g*. Description of a uniform system of distinctive pipe coloring proposed for use in Germany.

ABSTRACTS OF ARTICLES

UNTERSUCHUNGEN AN VENTILATOREN, A. Johnen. *Zeits. für Elektrotechnik und Maschinenbau*, November 30. The following data were obtained from tests made on three fans, two of the mine type with 2 m. (6.56 ft.) and 1.40 m. (4.59 ft.) diameter of the impeller; and one a forge fan 0.50 m. (1.64 ft.) diameter of impeller:

Tests of the fan having the radius of the impeller 1 *m.* (3.2 *ft.*). The fan had the following dimensions: Outside radius of the impeller 1 m. (3.28 ft.), inside radius $r = 0.60$ m. (1.96 ft.), radial length of the vanes 0.40 m. (1.31 ft.), the angle enclosed by the outside element of the blade with the tangent to the circle at that point 135 deg., the inside blade angle at the suction pipe 45 deg. The area of a vertical directing surface of a vane is $f = 0.144$ qm. (1.55 sq. ft.), the number of r.p.s. of the impeller 4, and, therefore, the volume of air enclosed by the impeller is $Q_0 = 2\pi [\frac{1}{2}(R + r)] fn = 6.28 \times 0.80 \times 0.144 \times 4 = 2.894$ cu. m. $= 102.1$ cu. ft. The theoretical loss of pressure is $\frac{\delta}{g} v^2$ where δ is 1.2 (average weight of air), g is the acceleration of a freely falling body in meters (9.81), and v for the given ventilator, at 240 r.p.m., is found to be 631, so that h_0 corresponds to a manometer pressure of 77.2 mm. (3.03 in.) of water. In Table 1 all the values are reduced to $n = 240$ r.p.m. For the normal section F of the suction port $= 1.06$ qm. (11.4 sq. ft.), the output of the steam engine was determined by indicator as $N_1 = 48$ h.p., no-load $N_0 = 15$ h.p., so that the fan took $N_u = 48 - 15 = 33$ h.p. The power in the motion of the air

$$N_n = \frac{Qh}{75} = \frac{22.520 \times 65.2}{75} = 19.58 \text{ h.p.}$$

and the efficiency of the fan referred to the steam engine is

$$\mu = \frac{N_n}{N_1} = \frac{19.58}{48} = 0.41,$$

and the air efficiency of the fan is

$$\eta = \frac{N_n}{N_u} = \frac{19.58}{33} = 0.59.$$

Table 2 gives indicator values with a ratio of transmission between the steam engine and ventilator as 1 : 6.

According to the investigations of Tresca, the ratio between the work used by the fan expressed in meter-kilograms and the third power of the number of revolutions of the driving engine, is approximately a constant, equal to about 0.05. The author gives a table containing 11 values of Tresca's ratio, all in the neighborhood of 0.05.

Mine fan with radius of impeller $R = 0.70$ *m.* (2.245 *ft.*). With $n = 400$ r.p.m., a peripheral velocity $v = 29.3$ m. per sec. (96 ft. per sec.), theoretical manometer pressure $h_0 = 105$ mm. (3.45 in.) of water and theoretical

TABLE 1 MINE VENTILATOR WITH RADIUS OF IMPELLER 1m. = 3.2 ft.

GENERAL DATA OF TESTS

Section of Suction Port F		Air Volume Q per Sec.		Manometer h in Water		Ratio of Pressures $h:h_0$	Ratio of Volumes $Q:Q_0$
Qm.	Sq. Ft.	Cu. M.	Cu. Ft.	Mm.	In.		
0.17	1.86	3.100	109.2	47.1	.185	0.61	1.07
0.63	6.78	12.146	428.	54.2	.213	0.70	4.20
0.83	8.93	17.280	610.	63.2	.248	0.82	5.97
0.97	10.41	21.678	763.	72.0	.283	0.93	7.49
1.06	11.39	22.520	795.	65.2	.256	0.84	7.78
1.42	15.25	26.921	950.	51.9	.204	0.67	9.30
1.87	20.01	33.590	1175.	46.9	.184	0.61	11.60
2.73	29.4	39.110	1380.	29.6	.116	0.38	13.51

TABLE 2 MINE VENTILATOR WITH RADIUS OF IMPELLER 1 m. = 3.2 ft.

MECHANICAL AND AIR EFFICIENCY

Number of Revolutions of		Total Power N_i	No-Load Work of Steam Engine N_0	Power taken by Ventilator N_u
Engine n_1	Ventilator n			
30	180	26.12	7.72	18.40
32	192	30.20	8.60	21.60
34	204	36.16	8.87	27.29
36	216	40.26	9.60	30.66
38	228	47.30	10.53	36.77
40	240	54.93	11.60	43.32
42	252	60.81	12.67	48.14
44	264	70.59	13.87	56.72
46	276	79.04	15.00	64.04
48	288	89.90	16.51	73.75
50	300	100.93	17.63	83.30

TABLE 3 MINE FAN WITH RADIUS OF IMPELLER 0.70 m. = 2.245 ft.

GENERAL DATA OF TESTS

Section of Suction Port F		Air Volume Q per Sec.		Manometer h in Water		Ratio of Pressures $h:h_0$	Ratio of Volume $Q:Q_0$
Qm.	Sq. Ft.	Cu. M.	Cu. Ft.	Mm.	In.		
0.15	1.61	3.227	117.5	66.5	.262	0.63	2.17
0.47	5.06	11.615	409.5	89.2	.351	0.85	7.80
0.72	7.75	16.280	570.	73.4	.289	0.70	10.93
0.83	8.93	17.588	622.	65.0	.256	0.62	11.80
1.32	14.2	24.085	850.	47.8	.189	0.46	16.16
2.72	29.3	28.584	1006.	15.9	.062	0.15	19.18

output of air 1,490 cu. m. (42.6 cu. ft.) per sec. Table 3 shows the efficiency of this ventilator, the assumptions being the same as in Table 1.

HOLZWARTH-JUNGHANS GASTURBINE. *Stahl und Eisen*, December 7. The Holzwarth-Junghans gas turbine belongs to the type of explosion turbines with intermittently working explosion chamber. It consists (Fig. 2) of a number of explosion chambers, *A*, placed in a ring, and each provided with air-inlet and gas-inlet valves, a stop-valve on the way to the nozzle, and several high-tension sparking plugs, as well as expansion nozzle *B*, as close to the explosion chamber as possible, and rotor wheel *C*, with velocity staging. The cycle of work is as follows: In one explosion chamber after another the mixture is ignited; the stop-valve to the nozzle is opened by the pressure developed by the explosion, and the gases of combustion do their work in passing through the nozzle to the wheel. After the end of the expansion, the stop-valve to the nozzle is made to close slowly in order to allow the air entering through the air-inlet valve thoroughly to scavenge the

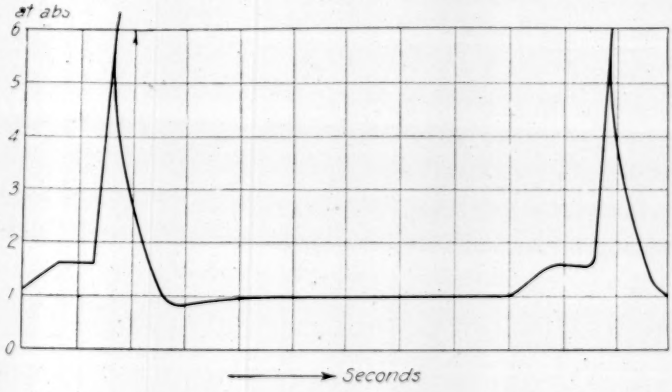

FIG. 1 HOLZWARTH-JUNGHANS GAS TURBINE INDICATOR DIAGRAM

combustion chamber, and effectively cool it as well as the nozzle and rotor wheel. As soon as the connection with the nozzle is cut off, the fuel is let into the combustion chamber, now filled with pure air, and the formation of eddies insures a thorough mixture. The cycle described is shown in the indicator-diagram (Fig. 1). The turbine constructed for experimental purposes for the inventors was of 1,000 h.p. (German, or 865 American h.p.), at 3,000 r.p.m., used power gas of 1,100 to 1,200 W.E. per cu. m. (124 to 135 B.t.u. per cu. ft.), and had 10 explosion chambers and a valve gear driven by the shaft *E*, receiving its power from the main shaft by a helical wheel transmission. The inventors have found that to have a "quiet" explosion which can be used with advantage in a "motor process," the combustion pressure must not exceed that which produces self-ignition, which is possible only with low initial temperatures.

TRAITEMENTS THERMIQUES ET MÉCANIQUES DES MÉTAUX À L'ATELIER, F. Robin and P. Gartner. *Revue de mécanique*, November 30. Annealing consists in heating a metal to a high temperature, but below that of fusion,

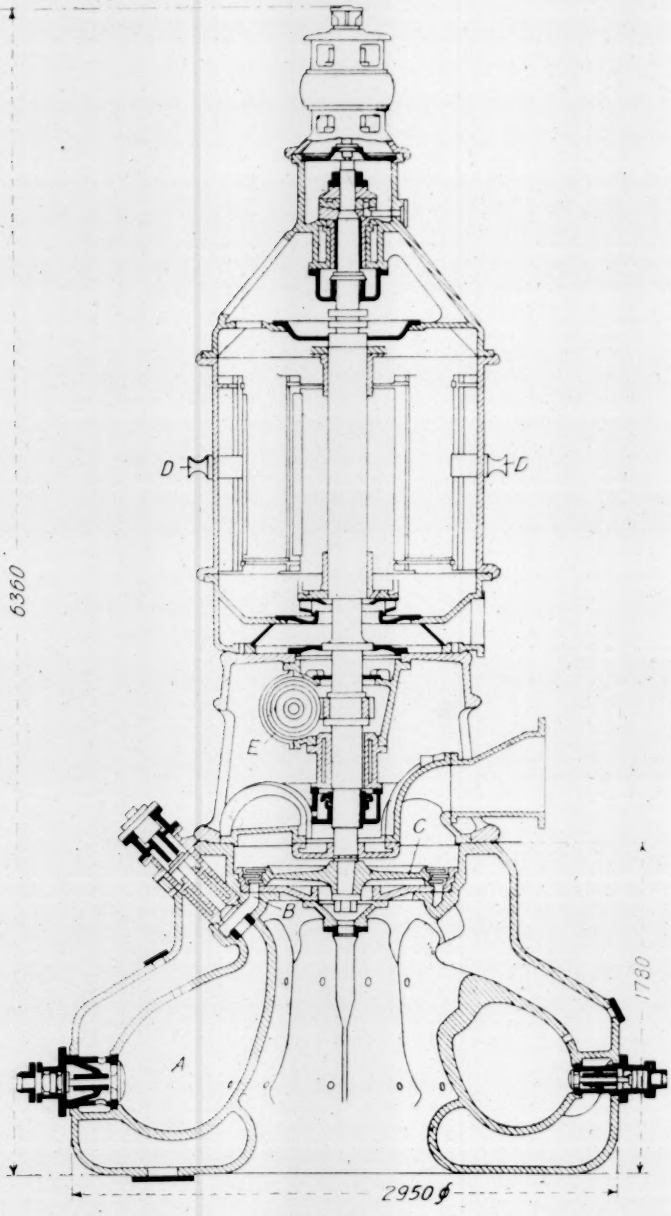

Fig. 2 Holzwarth-Junghans Gas Turbine

and then cooling it slowly down to the ordinary temperature so that it would attain its state of equilibrium and the mechanical properties belonging thereto. Annealing makes the constitution of the metal stable when cold; removes entirely the temper of the metal, which is the ability to maintain at ordinary temperatures a structure which is stable when the metal is hot; removes also the results of mechanical treatment, such as cold or hot hammering, and is often used to improve the qualities of a metal altered by heat or mechanical treatment which result in dangerous internal stresses, or too great crystallization.

Annealing of plastically deformed metal. A metal is said to be plastically deformed if it has undergone permanent deformations under a mechanical strain, which can be produced by forging, cold hammering, rolling. Hot plastic deformation, such as forging at 1,000 to 600 deg. cent. (1,832 to 1,112 deg. fahr.) for steel, 750 to 650 deg. cent. (1,382 to 1,202 deg. fahr.) for brass, changes the properties of metals less than cold, partly because hot metal lends itself better to internal changes generally, partly because subsequent cooling acts as annealing. Plastic deformation is not necessarily accompanied by visible changes in the structure; the strains break the crystals or deform them, and besides produce a sort of decrystallization which leads to changes in the properties of the metal, but the appearance of the elements does not change at all, and fine grain shown by microscope does not always indicate plastic deformation, because it could have been produced otherwise. Plastic deformation changes the chemical properties: drawn steel, brass and aluminum alloys after being subjected to it are more easily attacked by chemical agents than before; the heats of solution of steel in acids are also somewhat changed. It changes very materially the physical properties of metals, and increases the tensile strength as well as their limit of elasticity which in plastically deformed metal is inversely proportional to the elasticity in annealed metal. Hardness increases at the same rate as tensile strength, while the elongations in the tension tests fall off very rapidly, the malleability of the metal disappears, and its fragility increases. These variations are not proportional to the deformations; they grow rapidly at first for small deformations, then become slower and finally stop. At a comparatively slight deformation the limit of plastic deformation is reached, and the properties of the metal remain constant after that, no matter what the subsequent deformation may be. Impurities in metals raise the limit of plastic deformation quite noticeably; its limit varies with the temperature, and is generally lower at 500 or 600 deg. cent. (932 or 1,112 deg. fahr.) than at ordinary temperature.

Annealing brings the metal back to its normal state. In general, it acts gradually; the decrease of internal stresses is very slow at low temperatures, and it seems even that in some cases annealing during several minutes at 100 or 200 deg. cent. (212 or 392 deg. fahr.) makes even more prominent the effects of deformations. Then the decrease of plastic deformation goes on rapidly at first, but, as the annealing continues, it becomes slower, and seems limited to certain intervals of temperature. Finally, there is a zone of temperature at which crystallization by annealing is produced which grows with time and results in lower mechanical strength, in particular in

decreases of elongations and reductions of area. This crystallization is facilitated by the presence of impurities or previous permanent set, though it may be produced without it, by the metal being subjected to a high temperature for a long time alone. Practical conclusions: To be complete, annealing must be between certain limits of temperature. A strongly plastically deformed metal, if left for some time alone, tends to recover and to lose its special properties; this is what is called spontaneous annealing, and recent investigations show that contact with annealed metal accelerates the return to equilibrium in plastically deformed metal, all that making such metal very unstable and unfit for service, especially in punched and stamped pieces.

If the annealed alloy possesses a point of transformation corresponding to a change in constitution, the metal ought to be heated a little above it, and then cooled slowly, the time of its being kept at the high temperature being rather short; if the alloy possesses no point of transformation, it is heated to the tempering temperature and slowly cooled. The following temperatures may be recommended for practical purposes:

	Deg. Cent.	Deg. Fahr.
Copper	650	1,202
Nickel	500	932
Zinc	150	302
Aluminum	450	842
Silver and gold	650	1,202

Influence of annealing on the grain of steel. If steel is heated up to its highest point of transformation, its original crystallization disappears, the new grain is the finest the metal can take for the given chemical composition, and the metal keeps this grain after cooling; with increase of carbon and manganese in steel the best temperature for annealing is lowered, together with the points of transformation. If steel is heated for the usual length of time at a temperature below the point of transformation, the size of the grains does not change. If the temperature rises substantially above the highest point of transformation, the structure becomes clearly crystalline, of a permanent character, and the higher the temperature at the beginning of the cooling the larger are the grains. At each temperature a certain time is necessary for the grains to attain their maximum size. Slow cooling allows the grains to grow, especially at the beginning; as a result, the higher the temperature of annealing and the slower the cooling, the larger is the grain. Impurities affect the size of the grain; phosphorus, especially in low carbon steels, makes them grow, while nickel, chrome and vanadium act the other way. With permanently set steel the action of annealing begins at about 400 deg. cent. (752 deg fahr.), and increases with the rise in temperature and length of heating. Complete annealing begins at the point of transformation between 750 and 800 deg. cent. (1,382 and 1,472 deg. fahr.), and continues to about 1,000 deg. cent. (1,832 deg. fahr.), after which crystallization begins. For iron and soft steel the temperature must be about 950 deg. cent. (1,742 deg. fahr.), for medium hard steel 850 deg. cent. (1,562 deg. fahr.), and for hard steel about 800 deg. cent. (1,472 deg. fahr.); time 20 to 30 min.

LA RÉGULATION D'UNE TURBINE ALIMENTÉE EN VAPEUR À HAUTE ET BASSE PRESSION PAR UN ACCUMULATEUR HARLÉ. *Le Génie Civil*, December 2. An interesting case of the use of a steam accumulator in connection with low-pressure turbines taking steam from high-pressure engines working intermittently, *e.g.*, in rolling mills, is when a turbine has to work simultaneously with high-pressure steam from a boiler, and low-pressure steam received from another engine through an accumulator. Fig. 3 shows the arrangement of the parts when a Harlé accumulator is used.

The inverted tank *A* of the accumulator moving in an annular body of water *K* is provided with a fixed cam disc *B* which operates, by means of the cam roller *b* and rod *t*, the two valves *C* and *C'*: one on the pipe *D*

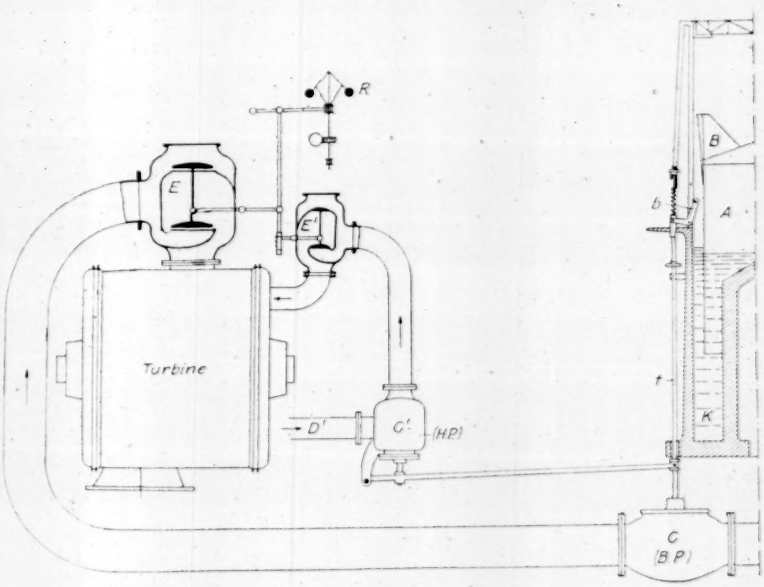

FIG. 3 TURBINE USING HIGH AND LOW-PRESSURE STEAM WITH THE FLOW OF STEAM REGULATED BY A HARLÉ ACCUMULATOR

carrying low-pressure steam from the accumulator, the other on the pipe *D'* coming from the boiler, the valves being so interconnected that when one opens the other closes. The two admission valves *E* and *E'* are controlled by the governor *R*, and arranged so that *E'* begins to open only when *E* is already fully opened, in order that the high-pressure steam should act only as supplementary to the low pressure.

EINE NEUE UEBERHITZERKONSTRUKTION. *Der praktische Maschinenkonstrukteur*, December 7. The fundamental conditions which a good superheater must satisfy, in the author's opinion, are a uniform distribution of steam over all the tube sections, and a sufficient division of the steam

into fine jets. The greater the state of division of the wet steam in the superheater tubes, the better it takes up heat through the tube walls which are heated by the gases on the outside. The superheater shown in Figs. 4 and 5, is made of straight or bent tubes *b*, and has its admission collecting tubes *a* provided with nozzles cc_1, the purpose of which is to divide the steam entering into the superheater tubes *b*, *i.e.*, to give to it two differ-

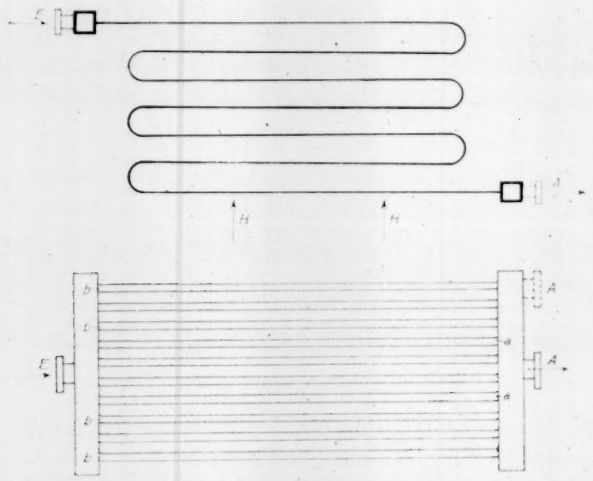

FIG. 4 PAUL KOCH SUPERHEATER

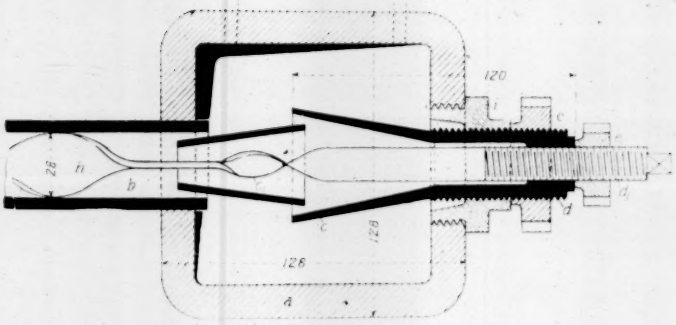

FIG. 5 PAUL KOCH SUPERHEATER SHOWING ARRANGEMENT OF NOZZLES

ent velocities. The arrangement of the nozzles can be such that the steam in one case passes close to the walls of the tube, and in another flows more along its axis, and thus breaks up the stream of steam with its water particles. The inventor of this superheater claims that, for a given

area, it is 50 per cent more efficient than a superheater without such an arrangement of tubes and nozzles.

Ueber Methoden zur Verhütung und Entfernung des Kesseleinsatzes, Willy Dahse. *Zeits. für Dampfkessel und Maschinenbetrieb*, December 8 and 22. This article presents a summary of recipes for the prevention of scale in boilers patented in Germany at various times. The methods used are partly mechanical, partly chemical. In the majority of cases the latter are more efficient. The following methods are mechanical: *Bernhard's* process: 1 kg. (2.2 lb.) of soap is mixed thoroughly with an equal quantity of gall and potato starch at a high temperature, paste or glue being used as a cohesive. This boiler composition may be used in the form of tablets or of a liquid. The lubricating properties of the composition insure the preservation of surfaces coming in contact with the steam. Unruh adds to the boiler feedwater some potato amniotic liquor about 50 l. per 10,000 l. of water (½ of 1 per cent). A certain amount of muddy sediment is formed at the bottom which must be let off from time to time. To get rid of scale already formed *E. Marris* uses a disincrustant consisting of starch and paraffin or wax. The mixture is spread over the dry scale in a layer about 1 cm. thick (a little over $^1/_{32}$ in.); the boiler is filled with water, and run for some days, during which time the scale breaks off in pieces. The starch is put in the mixture to prevent molten wax or paraffin and their gases from coming in contact with the surface of the boiler. After a few days the water is let out and the scale removed. *O. Lauffer* recommends for the prevention of formation and removal of scale the following composition: To 100 kg. (220 lb.) of tannin extract is added 1 kg. (2.2 lb.) of 25 per cent solution of ammoniacal salt, or of ammonium carbonate of equal concentration; this mixture is passed through cloth, cleared in filter presses, and brought up to density of 3 deg. Bé (1,023 American scale). It is kept at the temperature of 30 deg. cent. (86 deg. fahr.), and some rotting cheese is added to it, producing fermentation, and thereby eliminating all saccharine and oxidized products; there remain only combinations of tannin, very rich in hydroxyls, and efficient in the destruction of scale. The process of *Rüterbush* and *Bodstein* is based on the application of a petroleum emulsion: Grated quillaja bark or roots of common soapwort is moistened in a kettle by 30 per cent alcohol, left for 12 hours, and then diluted by 10 times its volume of water and heated. Solid particles are then separated out by filtration, and about 10 per cent of some inspissating substance, such as glycerin, starch or sugar, is added. Finally 10 parts of petroleum are added, little by little, in a thin stream, while the liquid is constantly stirred, and then the whole is left to cool completely. This emulsion is easily soluble in water, and can be kept for a long time without decomposition. *Smit* recommends for scale prevention a wash consisting of 1 kg. fish oil, 1 kg. horse fat, 1 kg. paraffin, 1 kg. white of zinc, 1½ l. (1.58 qt.) rain-water, 4 kg. graphite, 1 kg. lamp-black and 0.1 l. (0.1 qt.) carbolic acid. *Langrehr* suggests a wash consisting of 1 kg. graphite, 2 kg. milk and 20 kg. carbolic acid. *Kuhnert* claims to be able to prevent formation of hard scale by adding to the water ½ to 1·3 kg. (1.1 to 0.7 lb.) pumicestone for each qm. (10.76 sq. ft.) of water cov-

ered heating surface. *Becker* uses for the same purpose a mixture of 8 per cent ground pumicestone, 5 per cent powdered aluminum bronze and 87 per cent graphite. This mixture is said not only to prevent the formation of scale, and to remove scale already formed, but to remain in the boiler for a very long time, and to produce a polish on the boiler plates.

The following are chemical methods: *Doremus* introduces into the boiler potassium or sodium fluoride, hydrofluoric acid or cryolite and thus changes the scale into insoluble fluorides which form a precipitate, and can be removed by filtration. *Baudet* uses for the same purpose a mixture of 15 parts sodium hyposulphate, 10 parts rain-water, and 10 parts glycerin. *Schröder* adds sodium carbonate and magnesium oxide or hydroxide. *Bohlig* uses magnesium oxide or basic magnesium carbonate, and his method is said to be practical. *Bellanger* throws in a vessel, containing some 2 cu. m. (about 70 cu. ft.) water, 100 kg. (220 lb.) of hydraulic mortar or half that much cement, and the water is stirred until all the solid particles are suspended. Cement must be thrown in slowly. The liquid is then allowed to subside, and when the water has become clear it is decanted and the vessel, under constant stirring, is filled with fresh water. This is repeated until the hardness of the decanted water approaches that of fresh water, and then the deposit is dried and stored in sacks. This process frees it from soluble calcium salts. *Frank* adds to the water soluble soda glass until an alkaline reaction sets in, and then aluminum silico-fluorate up to a neutral or weak acid reaction. Soluble glass deposits the soluble salts of calcium, magnesium and iron, and aluminum silico-fluorate does the same for the cryoliths, etc. *Nieske* found that the addition of chrome salts prevents the formation of scale because the bicarbonates or sulphate of calcium pass into chrome salts, and easily settle down, leaving the boiler walls very clean.

UNTERSUCHUNGEN ÜBER DAS ALLGEMEINE VERHALTEN DES GESCHWINDIGKEITS-KOEFFIZIENTEN VON LEITVORRICHTUNGEN DES PRAKTISCHEN DAMPFTURBINENBAUES, Paul Christlein. *Zeits. des Vereines deutscher Ingenieure*, December 16. It was pointed out lately that we know comparatively little about losses in different parts of the steam turbine, and cannot even answer the simple question, whether the losses increase or decrease with the increase of velocity. To investigate this, the author made tests in the mechanical laboratory of the Technical Institute at Charlottenburg.

In actual experience when steam flows out through nozzles or distributors it meets with various resistances which reduce its theoretical velocity w_o to the actual velocity of efflux w, so that $w = \Phi\, w_o$, where Φ is the coefficient of velocity, and is equal to $\frac{w}{w_o} = 9.8\, \frac{R}{G}\, \frac{1}{w_o}$, R being the reaction pressure in kg., and G the weight of the flowing-out medium in kg. per sec., w in m. per sec., and in the direction of the geometrical axis of the nozzle or distributor. Since w_o could be obtained from the theoretical expression $w_o = 91.53 \sqrt{i_1 - i_2} = 91.53 \sqrt{\lambda}$ (where i_1 is the total heat of the steam at the entrance to the nozzle, and i_2 is the heat after frictionless adiabatic expansion to the back pressure p_2), to determine Φ it is necessary to find

experimentally the values of R and G. The tests have given the following results:

Nozzle with obliquely cut end and free expansion (Fig. 6). Fig. 7 gives

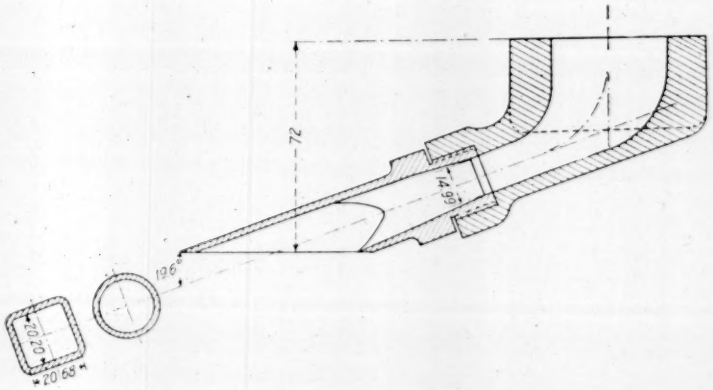

FIG. 6 NOZZLE WITH OBLIQUELY CUT END

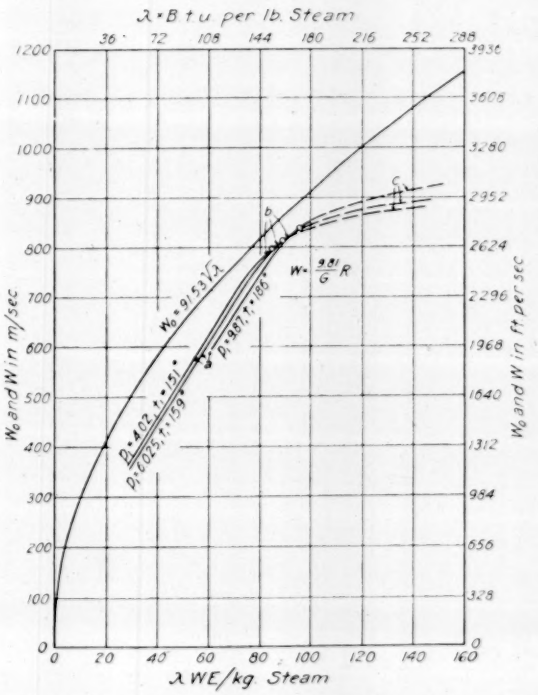

FIG. 7 CHARACTERISTIC OF THE NOZZLE SHOWN IN FIG. 6

as a function of the adiabatic loss of heat λ both the theoretical velocity

of efflux $w_o = 91.53\sqrt{\lambda}$, and the actual velocity of efflux $w = \frac{9.81\ R}{G}$ determined experimentally. The curve $w = f(\lambda)$ can be divided into three parts: (*a*) branch *a* rapidly rising from *o*, and representing the action of the nozzle at incomplete expansion, or back pressure $p_2^1 > p_2$, it being assumed that the widening of the nozzle is made to correspond to the ratio $\frac{p_2}{p_1}$

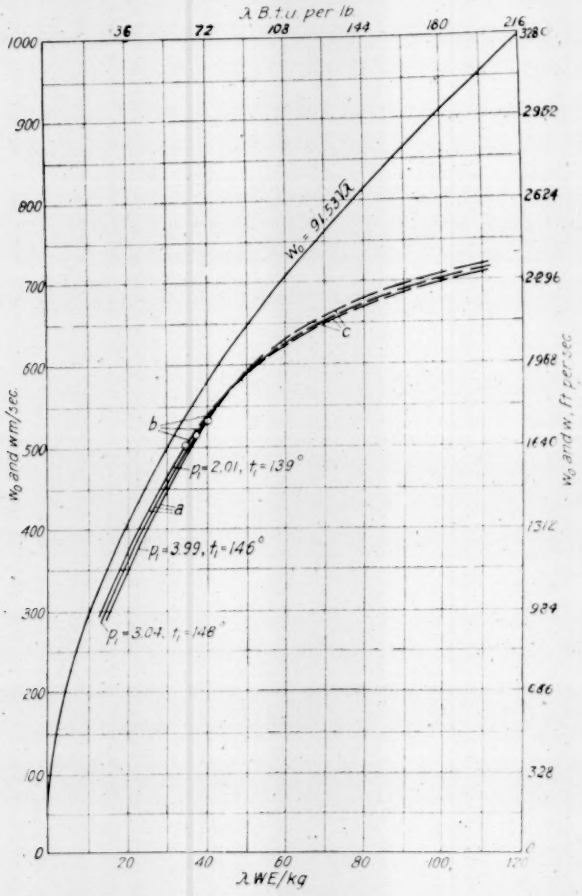

FIG. 8 CHARACTERISTIC OF A NOZZLE WITH PARALLEL WALLS

(*b*) the bend of the curve *b* for the action of the nozzle at complete expansion at $p_2^1 = p_2$, and in its immediate neighborhood; the curve *w* is very close to the theoretical velocity of efflux w_o, and the losses are very small, consisting of surface friction and internal friction between separate jets of the fluid; the efficiency of the transformation of energy, or the coefficient of velocity is here at its maximum. (*c*) the part *c* shows what hap-

pens when the widening of the nozzle is too slight, as compared with the ratio of pressures $\frac{p_2'}{p_1}$. The whole curve can be considered as the characteristic of a nozzle, and shows clearly for the first time how all the outside influences affect the transformation of energy in a given nozzle.

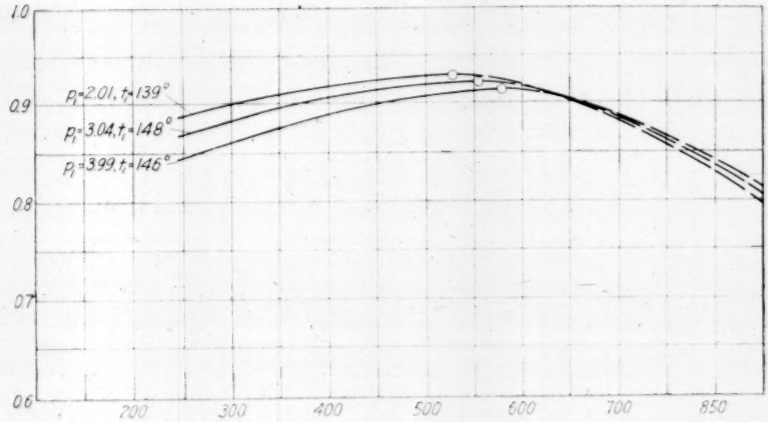

Fig. 9 Characteristic of the same Nozzle with Different Initial Pressures

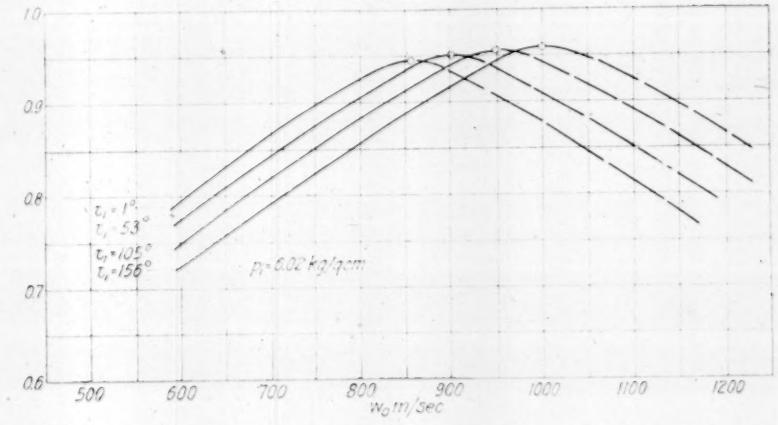

Fig. 10 Characteristic of the Nozzle shown in Fig. 6 with Different Degrees of Superheat

As a particular case of a nozzle with a conical widening, *i. e.*, $\frac{F}{F_m} > 1$, the case of $\frac{F}{F_m} = 1$ may be considered, that is the ordinary nozzle with parallel walls. Fig. 8 gives the $w = f(\lambda)$ curve for this case. The part *a* shows the influence of the surface friction against the walls of the nozzle, and of the internal friction between separate jets of the fluid; the part *b* where the curve lies in close proximity to the theoretical curve $w_a = f$

TABLE 4 DATA OF TESTS ON VELOCITY OF FLOW OF STEAM IN NOZZLES AND DISTRIBUTORS OF TURBINES

Angle of Lead, Deg. α	Axial Length, Mm.	Number of Passages	Material	Area up to the Smallest Cross-Section	Area from the Smallest Cross-Section to Outlet	Smallest Cross-Section of Nozzle ΣFm, Qcm.	Outlet Cross-Section ΣF, Qcm.	$\frac{\Sigma F}{\Sigma Fm}$	$\frac{d}{dm} = \sqrt{\frac{\Sigma F}{\Sigma Fm}}$	p_1 Kg/qcm	t_1 Deg. Cent.	t_1 Deg. Cent.
19.6	72	1	bronze	rough	smooth	1.766	4.15	2.35	1.533	4.02 6.025 9.870	151 159 186	8 1 7
20.1	63	1	bronze	rough	smooth	2.461	5.002	2.035	1.427	4.98 3.02 10.76	157 138 185	6 5 3
19.7	45	1	bronze	rough	smooth	0.5568	1.466	2.632	1.623	8.84 5.94 4.00	176 160 146	2 2 3
24.7	79	2	cast iron	rough	smooth	2.06	12.94	6.28	2.51	8.00 4.98	170 158	0.5 7
20.8	75	2	bronze	best machined	best machined	3.584	4.284	1.185	1.087	3.00 4.99	144 155	11 4
21.1	75	2	bronze	best machined	best machined	3.668	7.116	1.940	1.393	2.96	144	11
Test of Distributor												
21.6	75	3	cast iron with nickel-steel tips cast in	rough	rough	5.192	5.192	1	1	2.01 3.04 3.99	139 148 146	18 14 2

(λ) shows the working of the nozzle at a velocity exceeding, within definite limits, the so-called critical velocity; the part c shows what happens when the velocity exceeds the critical velocity beyond stated limits, and a considerable expansion in free space occurs after the fluid leaves the outlet of the nozzle.

In Fig. 9 the theoretical velocities w_o are used as abscissae, and the coefficients of velocities as ordinates. It shows the remarkable fact that *the efficiency of the transformation of energy in a nozzle is higher after the so-called critical velocity is exceeded*, contrary to now existing views.

Influence of superheat. The maximum coefficient of velocity is improved by superheating, all other conditions remaining as before, up to about 4 per cent, in proportion to the degree of superheating, as shown in Fig. 10. The author claims that as regards the maximum coefficient of velocity ϕ_{max}, apart from small (0.5 per cent) variations due to changes of initial pressure, it is increased from 1 to 2 per cent with increasing velocity w_o, the only exception to this being very small nozzles with a diameter of about 8 mm. (0.314 in.). The main point to which the author calls attention is that under correct conditions of work (*i.e.*, with a ratio of widening $\frac{F}{F_m}$ corresponding to the usual ratio of pressures $\frac{p_2}{p_1}$) the flow losses in the nozzle, all other conditions remaining unchanged, gradually decrease as the velocity increases. Table 4 gives the data obtained in the experiments.

OPYTY NAD VYRYVANIEM SVIAZEI I VYPOOCHIVANIEM PLOSKIKH STIENOK (Experiments on tearing-out of bolts and flexure of flat boiler walls), M. V. Gololobove. *Bulletin of the Permanent Committee of the Conferences of Representatives of Russian Railroads*, No. 10. Fig. 11 shows the arrangement used for these experiments. A copper sheet A (180 mm. x 180 mm. 7.1 in. x 7.1 in.) with a staybolt B wrapped in it was gripped in an iron frame made of two parts C and D. The upper part D was made of a flat sheet, 19mm. (0.75 in.) thick, with a square hole of 100 mm. (3.937 in.), equal to the average surface of a section of the furnace wall sustained by one stay. The two parts C and D were bolted together by four bolts E. The lower part C was provided with a special head to be placed into the grips of the testing machine; in the other pair of grips of the machine was placed the end of the staybolt B. This left a free square, 100 mm. (3.973 in.) on each side, out of which the testing machine tore out the stay. For these experiments sheets were taken 14, 10, 8 and 6 mm. (0.5, 0.39, 0.315 and 0.235 in.) thick, each with four threaded staybolts, out of which one staybolt had no head, and the rest had riveted heads of the following dimensions: (*a*) $h = 8$, $d = 42$ mm. ($h = 0.315$, $d = 1.65$ in.); (*b*) $h = 5$, $d = 37$ ($h = 0.195$, $d = 1.45$); (*c*) $h = 3$, $d = 31$ ($h = 0.118$, $d = 1.24$ in.). The sheet 14 mm. (0.5 in.) thick had three unthreaded staybolts with heads, and two threaded—one without head, and one with a normal head ($h = 8$, $d = 42$). This was done in order to determine experimentally the resistance to shearing offered by the head alone, when there is no threading to interfere. All the staybolts had the same diameters, 25 mm. (0.957 in.) in the threaded, and 22 mm. (0.865 in.) in the smooth part. The tables and curves give the results obtained in these experiments. The

article is to be continued (main results of these experiments will be given with the second part).

Einheitsfarben zur Kennzeichnung von Rohrleitungen in industriellen Betrieben. *Stahl und Eisen*, November 30. (The same article is printed in the *Zeits. des Vereines deutscher Ingenieure*, December 2). Several important German societies (Society of German Engineers, Association of Mining Interests in the Upper Mining Districts of Dortmund, Association of German Central Heating Concerns and the Society of German Engineers-Inspectors) have worked out a uniform system of distinctive coloring for pipes in industrial plants. But few colors are used in

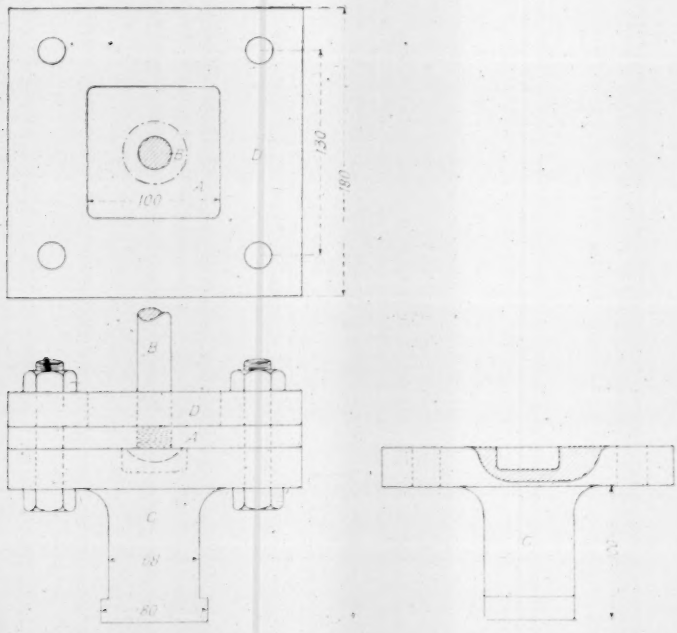

Fig. 11. Staybolt and Plate in the Experiment on the Tearing-Out of Staybolts

order that the men can easily remember them; there is no time to consult a table of colors in case of accident. The colors must be clear and easily distinguishable. The following basic colors are proposed: Green for water, yellow for gas, blue for air, white for steam, black for tar, rose for lye, brown for oil, gray for vacuum, rose with red streaks for acids. A fuller indication of the contents of the pipes, pressure, temperature, impurities or admixtures is effected by a combination of the basic colors with strips of the following colors: Red stands for danger, black for impurity, white for presence of steam, green for presence of water. The direction of flow may be indicated, if necessary, by arrows. The same colors may be used in drawings, steam being indicated by fine double lines. The colors on the pipes should be indicated by securely attaching to the pipes small painted and enameled or varnished strips of iron 3 in. to 4½ in. wide.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XIV)

ARTICLES IN PERIODICALS [1]

AUSSTRÖMEN VON GASEN DURCH DÜSEN, DIE, Georg Hamel. *Stahl und Eisen*, November 11. 3 pp., 4 curves. *e*.

On the flow of gases from nozzles.

BITUMINOUS GAS PRODUCER, NEW. *The Iron Age*, December 14. 1¼ pp., 3 figs.

Details and construction and gas purification of the Nordenson type developed by the Standard Gas Power Co.

CRUDE OIL ENGINE, A NEW. *The Engineer* (London), December 1. 1 p., 3 figs. *bf*A.

Two-cycle crude oil engine involving hot-bulb ignition, fuel injection direct into cylinder and crank case compression of air only. The possibilities of the two-cycle are not used to obtain reversal of direction of motion of the engine itself, a reversing gear being fitted.

CRUDE OIL GAS PRODUCER, THE GRINE. *Power*, November 14. ⅔ p., 1 fig. *b*.

Describes an outfit supplying gas to a 100-h.p. engine running practically at full load, 24 hours per day. The replenishment of the coke bed in generator is about 1/80 lb. per b.h.p-hr.

DIESEL ENGINE, THE KIND-MARINE. *International Marine Engineering*, December. 1½ pp., 4 figs.

General description of a reversing Diesel engine made by P. Kind & Co., Turin, Italy.

DIESEL-MASCHINE DER MASCHINENFABRIK J. E. CRISTOPH A. G. AUF DER POSNER AUSSTELLUNG 1911; Ernest, Neuberg. *Die Gasmotorentechnik*, November.

On the J. E. Cristoph Machine Works Co.'s Diesel engine as exhibited at the exposition of 1911, in Posen, Germany.

ENGINE, A NEW GAS. *The Engineer* (London), December 8. ½ p., 1 fig. *bfB*.

Describes engines, 10 to 100 h.p., built by Ambrose Shardlow & Co., Ltd., Ealing Wks., Sheffield.

ENGINE, THE TWO-STROKE CYCLE, R. G. Wells. *Engineering*, November 3. 3⅓ pp., 14 figs. *abf*.

Describes the various types and their operation.

ENGINE, THE RAYNER TWO-STROKE. *Power*, November 7. 1 p., 2 figs. *b*.

An effort to produce a sleeve-valve engine without infringing on the Knight patents. Description and details of engine.

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[1] Opinions expressed are those of the reviewer not of the Society. Articles are classified as *a* comparative; *b* descriptive; *c* experimental; *d* historical; *e* mathematical; *f* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

ENGINES, THEIR DESIGN AND APPLICATION, GAS, E. N. Percy. *International Marine Engineering*, December. 2½ pp.

Article continued from November issue.

GAS-ENGINE EXPERIMENTS, MECHANICAL COUNTER AND ELECTRICAL RECORDER FOR, John Eustice. *Engineering*, November 3. 1 p., 2 figs. *b.*

GASOLIN AUS NATURGAS, DIE GEWINNUNG VON. *Journal für Gasbeleuchtung und Wasserversorgung*, November 18. 2 pp. *def.*

The production of gasolene from natural gas.

MARINE DIESEL ENGINE PRACTICE, SOME IMPRESSIONS OF CONTINENTAL. *The Engineer* (London), December 8, 15, 22, 29. 7½ pp., 23 figs. *abfA.*

Notes and description of large two-cycle marine oil engines built by Carels Frères (December 8); eight-cylinder, submarine engine built by Maschinenfabrik, Augsburg-Nurnberg (December 15); Sulzer six-cylinder marine engines for the Italian mailboat Romagna (December 22); Schneider & Co.'s Sabathe, four-cylinder submarine engines (December 29).

MARINE OIL ENGINE, THE JUNKERS. *Engineering*, November 24. 3 pp., 12 figs. *b.*

Description and details of operation of a 200-h.p. engine and arrangement of a 1600-h.p. marine installation.

MOTEURS À PETROLE. *Revue de mécanique*, October 31.

———Carburateurs, G. Richard. 10 pp., 20 figs.

New types of carburetors: Bailey, Belleville, Bollee, Carter, Tolberth, Daimler.

———Diesel moteurs. 12 pp., 22 figs., 2 tables. *A.*

Describes Diesel motors of various types: Normand, Cavels & Schneider, Sulzer, Blackstone & Carter, Schiemann.

———Mises en train, G. Richard. 2 pp., 4 figs.

Arrangements of reversing gas engines. Types: Daimler, Sulzer, Remington.

———Moteurs Diesel à bord du petroleur Vulcanus. 2 pp., 5 figs.

Diesel motor on board S.S. Vulcanus. See also *The Engineer* (London), January 6, 1911.

MOTEURS À PETROLE, *Revue de mécanique*, November 30.

———Carburateurs. 11 pp., 45 figs.

Review of latest patents on carburetors.

NEUERE ROHÖLMOTOREN, Ch. Pöhlmann. *Dinglers Polytechnisches Journal*, November 4. 5 pp., 11 figs. *A.*

Describes new types of crude oil motors.

PEAT, POWER FROM. *The Engineer* (London), December 8. 1 p., 3 figs., 2 tables. *bdfA.*

Describes a successful 400-h.p. gas producer power plant at the factory of H. Robb, Portadown, using peat as fuel. This plant consists of two 200-h.p. gas producers, coke scrubbers, tar extractors, sawdust scrubber, exhausters and expansion box. This is claimed to be the only direct commercial utilization of peat working daily in the British Isles.

PRODUCER GAS. *The Engineer* (London), December 15. ½ p., 2 tables. *befA.*

Removal of dust and tar from producer gas by the dry or atmospheric method and the wet or water cooled method.

ROHOELMOTOREN DIESEL, NEUERE, Ch. Pöhlmann. *Dinglers Polytechnisches Journal*, December 23. 7 pp., 6 figs., 2 tables, 5 curves.

New type of crude oil motors, especially Diesel type.

TEER ZUM BETRIEBE VON DIESEL MOTOREN, VERWENDUNG VON, W. Allner-Dessau. *Journal für Gasbeleuchtung*, October 28. 10 pp., 24 figs., 3 tables, 16 curves. *A.*

Use of coal tar as fuel for Diesel motors.

THERMO-DYNAMIQUE DES GAZ, CONTRIBUTION À LA, Charles Beer. *Revue Universelle des Mines de les Métallurgie*, September. 47 pp. *e.*

Thermodynamics of gas.

TURBINE, THE EFFICIENCY OF THE GAS. *Engineering*, November 24. ⅔ p. *bc.*

From a paper read before the German Society of Naval Architects.

TURBINE, THE GAS, H. Holzworth. *Engineering*, November 24. 1⅓ pp., 1 fig., 2 curves. *b*.

Translation of paper before the Schiffsbautechnische Gesellschaft, Berlin, November 23, 1911.

VERBRENNUNGSMOTOREN IN ST. PETERSBURG 1910, DIE INTERNATIONALE AUSSTELLUNG, Alexis Bikoff and Georg von Doepp. *Die Gasmotorentechnik*, November. 4 pp., 5 figs., 4 tables, 1 curve. *b*.

On various internal-combustion engines at the International Exposition of 1910 in St. Petersburg, Russia.

GENERAL NOTES

SOCIETY FOR THE ADVANCEMENT OF EDUCATION IN THE SOUTH

The Society for the Advancement of Education in the South held its annual meeting, December 27–29, 1911, at Atlanta, Ga. The purpose of this society is to secure more exact and extensive knowledge of educational conditions and problems; more efficient union of educational forces; and more aggressiveness in advancing the cause of education. To further these ends some of the greatest national educators accepted invitations to speak before the meeting. The program also included papers and exhibits on recent conceptions of educational efficiency, education for the development of Southern rural life, industrial and vocational education in the South, agricultural education, rural sanitation, education for home life, education of adults, educational work of women's clubs, standardization of college education, recent criticisms of colleges, state inspection of elementary schools, adaption of education for life, the modification of the college curriculum to meet the demands of rural life, etc. An attractive feature of the meeting was a number of art exhibits by art dealers and artists. It consisted of a large number of carefully selected reprints and plaster casts of the great masters by the leading art dealers of the country. The exhibit also contained a number of paintings by Southern artists who are especially interested in developing art in the South.

NEW ENGLAND WATERWORKS ASSOCIATION

The annual meeting of the New England Water Works Association was held at the Hotel Brunswick, Boston, Mass., January 10, 1912. There were reports by the following committees: To look after and keep track of legislation and other matters pertaining to the Conservation, development and utilization of the natural resources of the country; To prepare a standard specification for fire hydrants; On uniformity of hose and gatenuts and direction of opening; On water consumption and statistics relating thereto; To collect information as to low water yields of catchment areas in New England, and at their discretion outside of New England.

AMERICAN SOCIETY OF CIVIL ENGINEERS

The annual meeting of the American Society of Civil Engineers was held at the society house, New York, January 17 to 20, 1911, and the following papers were presented for discussion: Drainage and Foundations; Fillers for Brick and Block Pavements; Bituminous Surfaces; Uses of Bituminous Material in Penetration and Mixing Methods. The list of excursions was particularly interesting this year, including visits to the aqueduct shafts, the Grand Central Terminal improvement, subway work, Fourth Avenue, Brooklyn, Mount Prospect Laboratory, Brooklyn; and to the Bush Terminal, Brooklyn.

ENGINEERS SOCIETY OF PENNSYLVANIA

The Engineers Society of Pennsylvania held its annual meeting at Harrisburg,

Pa., January 15, 1912. The convention was given over principally to the reading and discussion of the annual reports of the standing committees, and to a number of proposed amendments to the society's constitution. The subjects with which the committee reports dealt included public structures and buildings, water supply and sanitation, light and power, roads and highways, railroads and waterways, industrial education, conservation, telegraph and telephone communication, mines and mine safety, mineral lands, legislation and sociological problems relating to the engineering field.

WESTERN SOCIETY OF ENGINEERS

The annual meeting of the Western Society of Engineers, held in Chicago, Ill., January 10, 1912, was preceded by an excursion to the McCormick works of the International Harvester Co. In addition to the usual addresses by the retiring and newly-elected presidents, W. F. M. Goss, Mem. Am. Soc. M. E., and Judge Marcus Kavanaugh gave addresses on Citizenship. On January 11, an inspection trip was made to the steel car shops at Pullman and the new bridge shops of the American Bridge Co. at Gary, Ind. In the evening, a smoker was given at the society rooms, during which E. B. Frost of the Yerkes Observatory, gave a popular lecture on an astronomical subject.

WESTERN ASSOCIATION OF ELECTRICAL INSPECTORS

The annual meeting of the Western Association of Electrical Inspectors was held at the Plankinton House, Milwaukee, Wis., January 23–25, 1912. The sessions included the address of the president, V. H. Tousley, and the reports of committees; an address on The Protection of Buildings Against Lightning, by H. B. Long; a general discussion by the members regarding the improvements which have been made in the regulation of electrical hazards during the past year; addresses by W. J. Canada on Grounding of Conductors and Conduit, and by W. G. Middleton on Electrolysis; Reminiscences of a Pioneer Electrical Inspector, Franklin H. Wentworth.

PERSONALS

Claude M. Barron has entered the employ of R. W. Cameron & Co., New York. He was until recently associated with R. M. Wiggin of Mexico City, Mex.

C. E. Beck, formerly manager of the Sargent Steam Meter Co. of Chicago, Ill., has been appointed agent for the De La Vergne Machine Co. with offices in Chicago, Ill.

B. A. Behrend has been awarded the John Scott Medal by the Franklin Institute, for his contribution to the advancement of the electrical industry.

A. W. K. Billings has become associated with the Barcelona Traction, Light and Power Co., Barcelona, Spain.

A. C. Einstein, for a number of years president of the King Electric Light Co., the St. Louis County Gas Co. and the Suburban Electric Light and Power Co., has been appointed general manager of the Union Electric Light and Power Co. of St. Louis.

John W. Gray has been appointed superintendent of machinery of the Newport News Shipbuilding and Dry Dock Co., Newport News, Va. He was until recently associated with the Philadelphia, Pa. office of this company.

Walter J. Jones, consulting engineer of New York, has been appointed engineer to the Electrical Commission of the City of Montreal, Quebec. He will have immediate supervision over the work of gathering data and laying out a complete municipal underground conduit system for that city.

F. V. McMullin has been appointed treasurer of the Pennsylvania Forge Co., Philadelphia, Pa. He was formerly associated with the United Engineering and Foundry Co., Pittsburgh, Pa.

R. S. de Mitkiewicz has resigned as sales engineer with the Motor Engineering and Sales Co. of New York, to accept a position in the truck department of the Peerless Motor Car Co. of New York.

Wm. H. Parish has accepted a position as commercial engineer with the Westmoreland Coal Co., Philadelphia, Pa. Mr. Parish was formerly connected with the National Carbon Co., Cleveland O., as assistant superintendent.

B. G. Slaughter, Jr., has assumed the duties of engineer in charge of construction of the Pennsylvania Salt Manufacturing Co., Philadelphia, Pa. He was formerly connected with the Tennessee Copper Co., Copperhill, Tenn., in the capacity of chief engineer and superintendent of construction.

Chas. W. Stone has been appointed to succeed the late C. D. Haskins as manager of the lighting department of the General Electric Co., Schenectady, N. Y.

Frank R. Wheeler, formerly associated with the C. H. Wheeler Manufacturing Co., Chicago, Ill., has formed with R. B. Guernsey, the firm of Guernsey & Wheeler, with offices in San Francisco and Los Angeles, Cal., representing several eastern manufacturers of machinery.

Geo. R. Wood, consulting electrical engineer of Pittsburgh, Pa., has accepted a position in the electrical department of Berwin-White Coal Mining Co., Philadelphia, Pa.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for the Bulletin must be in hand before the 12th of the month. The list of men available is made up of members of the Society, and these are on file in the Society office, together with names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

0140 Engineer with broad experience, who has had some special training in handling office help and with a knowledge of accounting, to represent chief of department and make important decisions in his absence, in reorganization of the "internal economy" department of large Eastern city. Apply through Am. Soc. M. E.

0141 Man with considerable technical knowledge and organizing ability to take charge of municipal work having to do with contractors for street cleaning, garbage removal, paving, etc. Salary $6000. Eastern location. Apply through Am. Soc. M. E.

0142 Methodist Mission in Nadiad, India, wants mechanical engineer capable of giving instruction in both iron and wood work, to take charge of large industrial school.

0143 Ohio concern desires technical graduate, 25 to 30 years of age, preferably married, having practical experience as an executive in charge of an engineering department of a manufacturing plant, or as assistant to such executive, acquainted with modern systematic methods of conducting such work. Familiarity with construction of steel industrial and mining cars, design and construction of commercial auto trucks.

MEN AVAILABLE

350 Technical graduate, Junior member; qualified by five years' plant engineering experience, and proved ability to get results from men and equipment; to handle big work in either executive or shop end of manufacturing concern; thorough study of scientific shop management system and cost-accounting.

351 Technical graduate, Middle West state university, student member; desires position as assistant in mechanical laboratory instruction, in some Western or Middle West university; available about the middle of June 1912. Will consider work in gas-engine testing or erecting.

352 Member, graduate M. E.; 15 years' experience, wants position in teach-

ing or research work in connection with mechanical subjects; thoroughly versed in design of boilers, engines, turbines, mill and mining machinery and complete plants; some experience in structural steel and teaching; competent organizer and administrator.

353 Stevens graduate, married; varied experience with representative concerns in lines of manufacture and mechanical engineering, at present connected with large concern manufacturing product of high intrinsic value, requiring greatest detail, working in lines of reorganization and improvements of methods comprising planning, standardization of operations, classification, re-arrangement, cataloging, symbolizing stock, ready to take up work in similar lines. Leaving present employer with his full consent.

354 Professor of design; age 44; with 18 years' teaching experience, desires a change of location.

355 Member, technical graduate; 12 years' experience as a designer, construction engineer, mechanical engineer, and superintendent of industrial plants, desires a change of position; location, Middle West.

356 Member, graduate M. E.; 16 years' experience in steel works construction and other lines; charge both in field and office of the construction of several large plants; several years and until recently, chief engineer for firm of consulting engineers. Open for engagement along similar lines or any general lines.

357 Member of long standing; 40 years of age; married; 29 years' experience as machinist and tool maker; foreman, draftsman, designer, and superintendent to date, on modern manufacturing methods, planning work, writing up operations and designing tools and fixtures. Capable of taking hold of any reasonable mechanical proposition, tooling it up and estimating cost of machinery and tools, output, etc.

358 Associate member wishes to change; technical education; 12 years' experience, familiar with latest manufacturing methods; desires position requiring extensive knowledge of the shop, such as master mechanic or efficiency engineer.

359 Member in Baltimore is available as sales engineer and upon a commission basis, for Maryland, Virginia, West Virginia, and contiguous territory; experienced with machinery of cement mills, lime plants, quarrying, crushing, grinding, conveying, etc.

360 Technical engineer; age 31; 4 years' shop experience; 1 year electrical testing department; 7 years on design and construction of copper reduction works, locomotive shop and railway repair shop, desires position as designer or field engineer.

361 Engineer, technically trained with 14 years' shop experience, designing, engineering sales, etc., now employed as general manager, desires to change for position in business or executive end of manufacturing concern.

362 Associate member; 20 years' experience in designing all kinds of steam, pumping, blowing, hoisting, electric light and gas engines, gate hoists, transmission machinery, and power plants; desires a position as chief draftsman or chief engineer.

363 Mechanical engineer specialized in boiler practice and industrial furnaces. Extensive experience in design, construction and operation of power and manufacturing plants, desires position with consulting engineer or gas engineer for power or industrial plant.

364 Mechanical engineer, Junior member, technical graduate, wide experience in steam engineering including economical operation of leading types of water and fire tube boilers, operation of high-speed steam, hoisting engines, pumps, conveying machinery. Experienced in erection of locomotives for leading railway and good knowledge of steel mill work. Desires position where above experience will be of value, or as assistant to the engineer, manager or superintendent in some manufacturing business where there are good chances for advancement.

365 Sales manager or assistant; connected with selling end for the last five years; can handle any type or class of machinery, or open for manager of some manufacturing concern to handle plant.

366 Technical graduate, married, 32 years of age, desires position as manager or superintendent of small growing factory; practical experience in machine shop and drafting room, devising and installing complete cost, shop and production systems, factory executive in responsible charge. Experienced in laying out and installing industrial plants and electric installations.

367 Junior member, mechanical engineer with college training, one and one-half years' experience in machine shops, two as junior salesman, one as technical editor; familiar with safety appliances for shop and factory.

368 Member of long experience as chief engineer, superintendent, manager, and executive head of industrial corporations; confidential investigations and advice on industrial propositions and corporations; special experience in determining commercial value and "fiting risks" of patents, plants and manufacturing propositions with best method for marketing or development.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary Am. Soc. M. E.

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A new edition of this well-known laboratory manual, of which over 10,000 copies have been sold. It has been largely rewritten, containing over 200 pp. more than the sixth edition. It is of particular value to the consulting engineer as it contains in a single volume the principal standard methods of testing which have been adopted by various engineering societies, and an extensive series of tables useful in computing results.

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GOLDSCHMIDT THERMIT CO., *New York.* Reactions, December 1911, a quarterly periodical devoted to the science of aluminothermics, 20 pp.

HYDRAULIC PRESS BRICK CO., *Kansas City, Mo.* Testimonials as to the efficiency of the Ray feedwater purifier, 8 pp.

INGERSOLL-RAND CO., *New York.* "Imperial" valveless telescope feed hammer drill, 16 pp.

NELSON VALVE CO., *Philadelphia, Pa.* New price lists of iron body Nelson gate valves, 28 pp.

PRATT & WHITNEY CO., *Hartford, Conn.* Vertical shaper, 7 pp.

RUSSELL & CO., *Massillon, Ohio.* Steam road locomotives, 33 pp.; gas tractors, 28 pp.; general utility traction engine, 32 pp.

VALLEY CITY MACHINE WORKS, *Grand Rapids, Mich.* Wood working machinery, 16 pp.

WESTINGHOUSE, CHURCH, KERR & CO., *New York.* Central power stations, 49 pp.

YARNALL-WARING CO., *Philadelphia, Pa.* The Blow-Off, December 1911, a trade publication, devoted to the interests of valve design, 12 pp.

UNITED ENGINEERING SOCIETY

AUTOBIOGRAPHY OF JOHN FRITZ. *New York, 1912.* Gift of John Fritz.

TRADE CATALOGUES

AMERICAN VANADIUM CO., *Pittsburgh, Pa.* Nine photographs of vanadium steel machine parts, 9 pp.

H. CHANNON CO., *Chicago, Ill.* Machinery and general supplies for railroads, manufacturing and power plants, 952 pp.

DU PONT DE NEMOURS POWDER CO., *Wilmington, Del.* Use of explosives on the farm, 118 pp.; dynamiting boulders and stumps, 28 pp.; tree planting with Du Pont dynamite, 24 pp.

FAY & EGAN CO., *Cincinnati, Ohio.* Woodworking machinery, 384 pp.

FORBES CO., *Philadelphia, Pa.* Water sterilization by heat, 80 pp.

MUNNING-LOEB CO., *Matawan, N. J.* Belt-driven and motor-driven electroplating dynamos, 19 pp.

TINIUS OLSEN & CO., *Philadelphia, Pa.* Pt. A, catalogue of testing machinery, 66 pp.; pt. B, spring testing machinery, 16 pp.; pt. C, cement testing apparatus, 19 pp.; pt. D, fiber, paper, glue, rubber testing machinery, 12 pp.; pt. E, wire, chain, anchor testing machinery, 20 pp.; pt. F, oil, grease, bearing metal testing machinery, 10 pp.; pt. G, transverse testing machinery, 8 pp.; pt. H, special testing machinery, 24 pp.

CHANGES IN MEMBERSHIP

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ALBRIDGE, Frederick Royal (1911), Ch. Draftsman, Bessemer & Lake Erie R. R., and *for mail*, care of H. H. Lininger, 16 Plum St., Greenville, Pa.

ATKINS, Harold B. (1903), Cons. Engr. and Certified Pub. Acct., 1409 West St. Bldg., and *for mail*, 527 W. 121st St., New York, N. Y.

BARNES, Charles B. (1905; 1908), Asst. Wks. Engr., West. Elec. Co., Hawthorne Sta., Chicago, and *for mail*, 229 Wisconsin Ave., Oak Park, Ill.

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BEECHER, J. F. (Associate, 1908), Checker Coke Oven Dept., Tenn. Coal, Iron & R. R. Co., and *for mail*, 2201 Fifth Ave., Birmingham, Ala.

BIRDSEY, Charles Robt. (Associate, 1907), Ch. Engr., U. S. Gypsum Co., 200 Monroe St., and *for mail*, 528 S. Millard Ave., Chicago, Ill.

BURCH, H. Kenyon (1910), Mech. Engr., 645 Bixel St., Los Angeles, Cal.

BURGOON, Charles Eli (1907), Burgoon-Matthews Elec. Co., 3 N. 20th St., Birmingham, Ala.

ESTABROOK, Mansfield (Junior, 1903), Niles-Bement-Pond Co., 111 Broadway, New York, N. Y.

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GRAY, John Wilson (Junior, 1895), Supt. of Mchy., Newport News Shipbuilding & Dry Dock Co., and *for mail*, 124 28th St., Newport News, Va.

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JACOBS, Ward S. (1897; 1904), Treas., The Walton Co., and *for mail*, 530 Farmington Ave., Hartford, Conn.

KING, Roy Stevenson (1904; 1910), 612 Barnard St., Savannah, Ga.

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HVID, Marinus (Junior, 1911), Designer, Blast Furnaces and Rolling Mill Equip., 7532 Champlain Ave., Chicago, Ill.

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NORTON, Homer Burdett (1911), Ch. Engr., Elk Tanning Co., and *for mail*, 302 Adams Ave., Ridgway, Pa.

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WALLACE, Erwin Van Doren (Junior, 1911), Teacher Mech. Drawing, Manual Training and Mathematics, Central High Sch., St. Louis, Mo.

PROMOTIONS

HARLAN, O. K. (1904; 1911), Mech. Draftsman, Box 65, Empire, Canal Zone, Panama, and 811 W. Oak St., Union City, Ind.

LEUTWILER, Oscar A. (1904; 1911), Asst. Prof. Mch. Design, Univ. of Ill., and *for mail*, 511 W. Green St., Urbana, Ill.

COMING MEETINGS

FEBRUARY–MARCH

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading, and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 15th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AMERICAN ASSOCIATION OF RAILROAD SUPERINTENDENTS
March 17, annual meeting, Chicago, Ill. Secy., O. G. Fetter, Carew Bldg., Cincinnati, Ohio.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
February 9, monthly meeting, 29 W. 39th St., New York. Secy., F. L. Hutchinson.

AMERICAN RAILWAY ENGINEERING ASSOCIATION
March 19–21, annual convention, Chicago, Ill. Secy., E. H. Fritch, Monadnock Blk.

AMERICAN SOCIETY OF CIVIL ENGINEERS
February 7, 21, bi-monthly meetings, 220 W. 57th St., New York. Secy., C. W. Hunt.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
Monthly meetings: St. Louis, February 7; New York, February 13; Boston, February 16. Secy., Calvin W. Rice, 29 W. 39th St., New York.

CANADIAN MINING INSTITUTE
March 6, annual meeting, Ottawa, Can. Secy., H. Mortimer-Lamb, Windsor Hotel, Montreal.

CEMENT PRODUCTS EXHIBIT
January 29–February 3, exhibit, New York. Office, 72 West Adams St., Chicago, Ill.

IOWA ENGINEERING SOCIETY
February 21–23, annual meeting, Davenport, Iowa. Secy., S. M. Woodward, Iowa City.

NATIONAL BRICK MANUFACTURERS ASSOCIATION
March 6–9, annual convention, Chicago, Ill. Secy., T. A. Randall, Indianapolis, Ind.

NEW ENGLAND ASSOCIATION OF GAS ENGINEERS
February 14–15, annual meeting, Boston, Mass. Secy., N. W. Gifford, 26 Central Sq., East Boston.

NEW ENGLAND RAILROAD CLUB

February 13, regular meeting, New American House, Hanover St., Boston, Mass. Paper: Mechanical Handling of Freight, W. C. Carr. Secy., G. H. Frazier, 10 Oliver St.

RAILWAY SIGNAL ASSOCIATION

March 18, semi-annual meeting, New York. Secy., C. C. Rosenberg, Bethlehem, Pa.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
February			
1	Blue Room Engineering Society	W. D. Sprague	8.00 p.m.
8	Illuminating Engineering Society	P. S. Millar	8.15 p.m.
9	American Institute of Electrical Engineers	F. L. Hutchinson	8.15 p.m.
13	American Society of Mechanical Engineers	C. W. Rice	8.15 p.m.
16	New York Railroad Club	H. D. Vought	8.15 p.m.
28	Municipal Engineers of New York	C. D. Pollock	8.15 p.m.
March			
5	Institute of Operating Engineers	H. E. Collins	8.00 p.m.
8	American Institute of Electrical Engineers	F. L. Hutchinson	8.15 p.m.
12	American Society of Mechanical Engineers	C. W. Rice	8.15 p.m.
14	Illuminating Engineering Society	P. S. Millar	8.15 p.m.
15	New York Railroad Club	H. D. Vought	8.15 p.m.
27	Municipal Engineers of New York	C. D. Pollock	8.15 p.m.
28	Institute of Operating Engineers	H. E. Collins	8.00 p.m.

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